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Main Building, Armour Institute of Technology.

CYCLOPEDIA OF DRAWING

MECHANICAL DRAWING

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WORKING DRAWINGS

MACHINE DESIGN

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SHEET METAL PATTERN DRAFTING

TINSMITHING

PRACTICAL PROBLEMS IN MENSURATION

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New York Trade School.

Compiled from the Instruction Papers *in the* Mechanical Engineering *and* Sheet Metal Pattern Drafting Courses *of the* American School of Correspondence *at* Armour Institute of Technology : : : : : : : : : : : : : : *Chicago, Illinois*

Edited by

ALFRED E. ZAPF, S. B.

Secretary American School of Correspondence

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THIS VOLUME CONSISTS OF EIGHT OF THE FIFTY REGULAR INSTRUCTION PAPERS IN THE MECHANICAL ENGINEERING AND SHEET METAL PATTERN DRAFTING COURSES OF THE AMERICAN SCHOOL OF CORRESPONDENCE, INDEXED AND BOUND TOGETHER IN CONVENIENT FORM FOR READY REFERENCE, BUT NOT IN THE ORDER USUALLY STUDIED.

THE INSTRUCTION PAPERS OF THE AMERICAN SCHOOL OF CORRESPONDENCE ARE PREPARED EXCLUSIVELY FOR THE USE OF ITS STUDENTS BY MEN OF ACKNOWLEDGED PROFESSIONAL STANDING, AND REPRESENT YEARS OF SPECIAL PREPARATION NECESSARY TO ADAPT THEM TO THE NEEDS OF PERSONS OBLIGED TO STUDY WITHOUT THE DIRECT ASSISTANCE OF A PERSONAL TEACHER.

THE CHIEF AIM OF THIS WORK IS TO ACQUAINT THE PUBLIC WITH THE STANDARD, SCOPE, AND PRACTICAL VALUE OF THESE PAPERS THROUGH AN OPPORTUNITY FOR PERSONAL EXAMINATION; AND IT IS HOPED THAT SUFFICIENT MATERIAL IS GIVEN HERE TO AROUSE IN THE READER A DESIRE TO KNOW MORE.

ALTHOUGH PUBLISHED PRIMARILY TO SHOW THE CHARACTER OF THE INSTRUCTION OFFERED BY THE AMERICAN SCHOOL OF CORRESPONDENCE, AND REPRESENTING ONLY A SMALL PORTION OF THE COMPLETE COURSE, IT IS CONFIDENTLY BELIEVED THAT THIS VOLUME HAS IN ITSELF ENOUGH CONDENSED, PRACTICAL INFORMATION TO MAKE IT OF IMMEDIATE VALUE TO THE DRAFTSMAN, STUDENT, OR TEACHER.

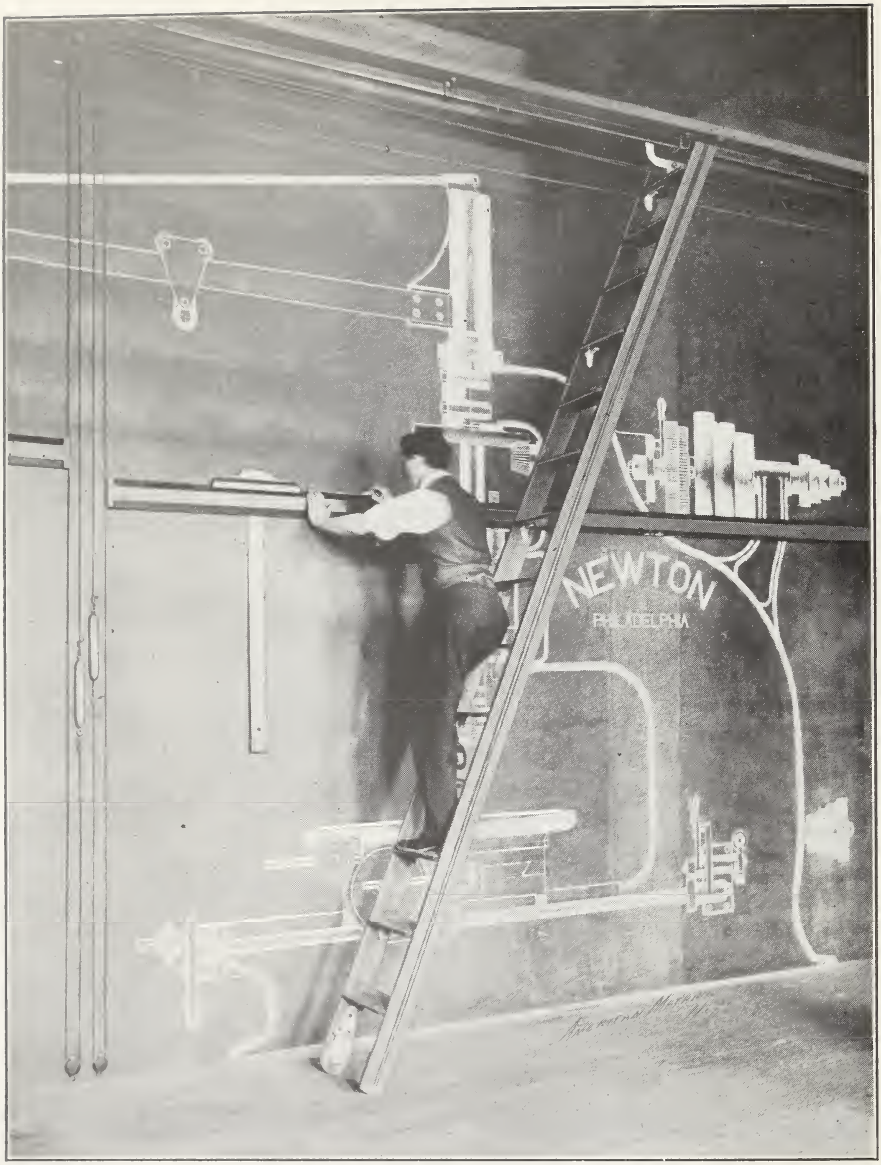
EXAMINATION QUESTIONS

FOLLOWING EACH SECTION ARE THE QUESTIONS OR PLATES WHICH CONSTITUTE THE REGULAR EXAMINATION OF THE AMERICAN SCHOOL OF CORRESPONDENCE. THEY OFFER THE READER A MEANS OF TESTING HIS KNOWLEDGE OF THE SUBJECTS TREATED.

INABILITY TO ANSWER THESE QUESTIONS, OR TO SOLVE THE PROBLEMS, WILL SERVE TO SHOW THE NECESSITY FOR FURTHER STUDY.

THE READER IS URGED TO SOLVE EVERY PROBLEM, CHECKING HIS RESULTS WHEREVER POSSIBLE WITH SIMILAR PROBLEMS IN THE PRECEDING PAGES. THIS WILL AFFORD AN EXCELLENT MEANS FOR FIXING THE MATTER IN HIS MIND.

STUDENTS PREPARING FOR COLLEGE OR CIVIL SERVICE EXAMINATIONS WILL FIND THESE QUESTIONS OF GREAT VALUE



LARGE BLACKBOARD USED IN THE DRAWING ROOM OF THE NEWTON MACHINE TOOL WORKS.

MECHANICAL DRAWING.

WORKING DRAWINGS.

In Mechanical Drawing Parts I, II and III, instruments and materials are described and some hints given regarding the use of compasses, line pen, triangles, T-square, etc. In addition, the general principles upon which all Mechanical Drawing depends, are explained. After completing this work the student should be able to draw neatly and accurately and apply the fundamental principles.

Let us now take up the subject of working drawings and see how the principles of orthographic projection are made of practical use. We shall see, as we go on, that to a great extent the theoretical principles already learned in the study of projections are used in practical working shop drawings. Nevertheless, there are certain instances in which actual practice differs slightly from the theory.

We shall also find that all draftsmen do not follow the same customs in the matter of minor details, but that in many cases there are several ways of representing objects, all of which may be equally correct, one draftsman using one method either because he prefers it or because it best serves the purpose for which his particular work is being done, while another draftsman uses a different method. The more important principles and customs, however, are pretty well established.

A study of the subject of working drawings should, first, teach us the methods of the best drafting rooms; second, should train our judgment to decide how best to represent the particular object which we have to draw; third, should train our hand and eye to make a clear, neat and well-executed drawing, without unnecessary expenditure of time.

Definition of Working Drawing. A working drawing of any object is a drawing which completely describes the object in every particular, showing its form, size, material, finish, and all other details, so that a workman may take the drawing and without any further instructions make the object exactly as the draftsman intended it to be made. The drawing is, therefore, a sort of language, by which the man who designs the object describes it to the man who is to make it. Fig. 1 shows the working drawing of a bell crank lever. The drawing itself shows the shape and the dimensions show the size.

Aside from drawings of buildings, etc., that is Architectural drawings, the greater part of the working drawings which are made are for machines, and we will consider chiefly the latter, or, as they are called, machine drawings. These drawings are almost always orthographic projections, as this is by far the easiest and best way to represent a machine or a part of a machine. Sometimes, when it is desired to give a sort of bird's-eye view of a machine, an isometric or an oblique projection is made, but this method involves so much labor that it is seldom used.

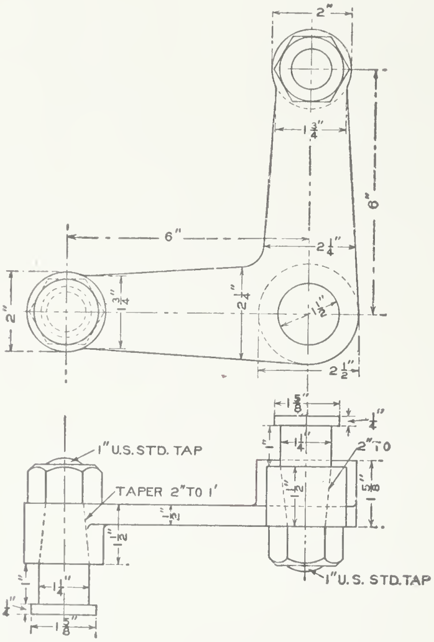


Fig. 1.

Lines. In order to make a drawing perfectly clear, and to avoid confusing one line with another, different kinds of lines are used for different purposes. Fig. 2 shows the six most common kinds used. The ordinary *full lines* are used to represent visible lines of the object which is being drawn.

The *invisible lines* are used to represent lines of the object which would be hidden from sight if a person looked in the direction in which he imagines himself to be looking while he is drawing it.

The use of the *shade lines* will be explained later.

Center lines are used to connect different views of an object, the line being drawn through the center of the piece and extending through both views. Locations of holes are usually shown by having two center lines drawn at right angles to each other through their centers, and the position of these center lines located. Wherever a dimension is to be given to the center of a piece a center line is drawn through the piece and the dimension given to this line.

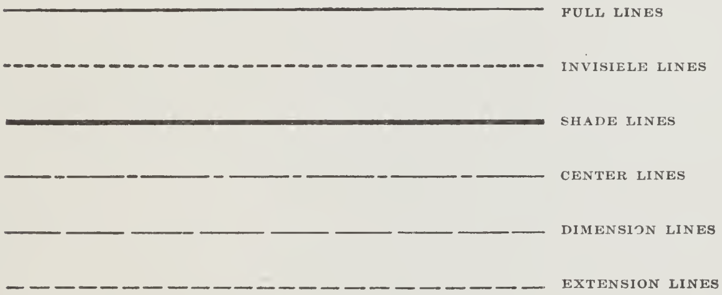


Fig. 2.

Extension lines are sometimes used to connect two views of a piece, but, wherever it is possible to use a center line, instead, the latter is preferable. The principal use of extension lines is, as the name implies, to extend the lines of the object so as to give dimensions between them.

Dimension lines are used in giving dimensions from one line or point to another.

The ordinary lines and the invisible lines should be made of the same width; the shade lines should be made considerably heavier, and the center lines, extension lines and dimension lines should be lighter.

Location of Views. In our preceding study of projections we imagined our object to be held in the angle formed by two

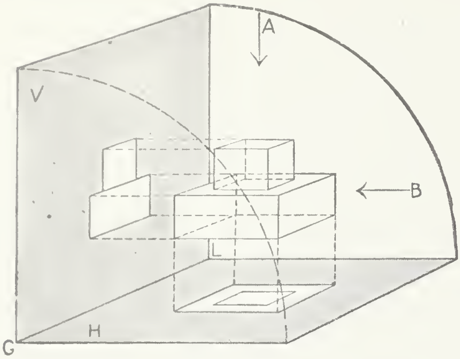


Fig. 3.

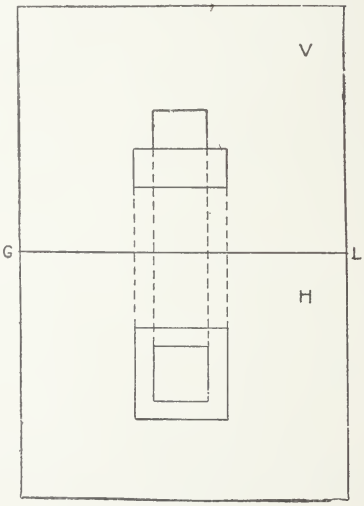


Fig. 4.

planes which intersect at right angles, the planes and the object being supposed to be in the position represented in Fig. 3, the horizontal plane H being below the object, and the vertical plane V being beside the object. We then projected down to the horizontal plane and represented there the object as it would appear if seen squarely in the direction indicated by arrow A. We next projected into the vertical plane, and represented the object as it would appear if the line of vision took the direction of the arrow B. Our planes, with the horizontal and vertical projections of the object on them, when laid out flat, as a sheet of paper is when we actually draw on it, would appear as in Fig. 4, the horizontal projection, or top view, being *underneath* the vertical projection, or side view. This is the practice followed by some draftsmen in making working drawings. Many of the best draftsmen, however, think that it is not as clear and convenient to have the top view of the object underneath the side view, but that it is better to have the view of the top above the other view. Consequently, it is becoming more and more a general custom to suppose the planes to be transparent and to be located with respect to the piece which is being drawn as shown in Fig. 5, the H plane being above the object and the V plane in front of the object. They then draw on the H plane the object as it appears when looking at it through the transparent plane in the direction indicated by arrow C, and on the V plane the object as it appears when looking in direction of arrow D. Now, when the two planes are laid out flat, the top view is above the side view, as in Fig. 6.

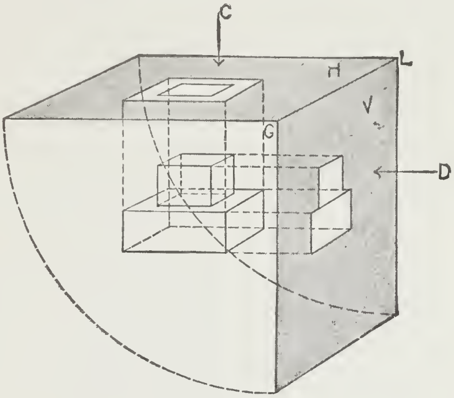


Fig. 5.

Another way of showing this is to suppose the object to be located in a transparent box, and that we look at it from the top and the various sides of the box, and draw on these sides the object as it appears from that side. Then, if the sides of the box are laid out flat, the side view will come in the middle, with the top view above, the view of the bottom underneath, the view of the right-hand side on the right, and the view of the left-hand side on the left.

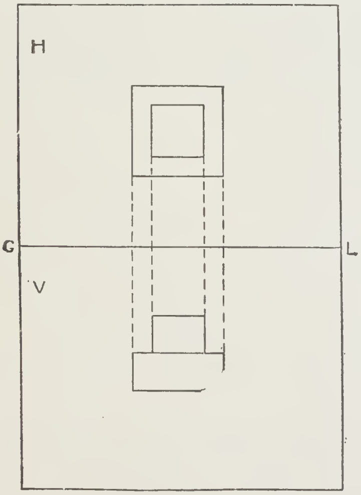


Fig. 6.

From now on, in our work on drawing, we shall follow this rule.

Cross-sections. Very often it occurs that a piece is hollow and the inside construction is more or less complicated, so that if in the drawing the outside view is shown, with the invisible interior drawn in dotted lines, the latter are so confused that the drawing is not easily understood. For this reason it is often convenient to imagine the piece to be cut open, and to draw it as if we were looking directly at the inside. Such a drawing of a piece is called a cross-section of the piece. The material which must be cut if the object were actually split open is then crosshatched, different kinds of crosshatching being used for different materials. Fig. 7 shows the kinds of cross-hatching in ordinary use.

Fig. 8 shows a plan and elevation of a simple piece with a hole through it, the hole being shown in the elevation by dotted lines. Fig. 9 shows a plan and a cross-section of the same piece, the cross-section being drawn as if we were looking in the same direction as in the elevation of Fig. 8, but in Fig. 9 the front half is supposed to be cut away and we are looking at the inside of the back half. It will be observed that even on a piece as simple as the one here shown, the shape of the interior is much clearer from the cross-section than from the elevation, and on a more complicated piece the same will be true to a greater extent.

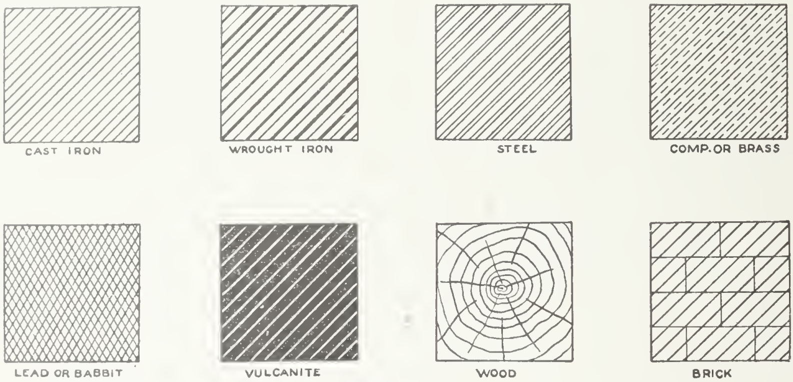


Fig. 7.

It should be borne in mind that ordinarily in making a cross-section we show not only those parts of the object which lie in the plane in which it is supposed to be cut, but also all which lie back of that plane.

Sometimes it is desirable to show the inside and the outside of a piece on the same view. Fig. 10 shows the same piece as in Fig. 8, but in Fig 10 the left-hand half of the lower view is an elevation and the right-hand half is a cross-section, the section and the elevation each extending to the center and being separated by a center line. This is as if one-quarter of it were cut away, the cuts being made along the lines C A and C B of the plan view.

Of course this combination of an elevation and cross-section can be used only when the right-hand and left-hand halves of the piece are alike.

Shade Lines. In order to make drawings easier to read, and to make the parts of the object stand out more clearly, shade lines are often put on the drawing. The general principle which determines what lines shall be shade lines is the same as that which governs shade lines already studied under the subject of projections. If, however, this theoretical principle were to be followed out exactly on drawings of machines, and other complicated drawings, it would involve a great deal of time and labor.

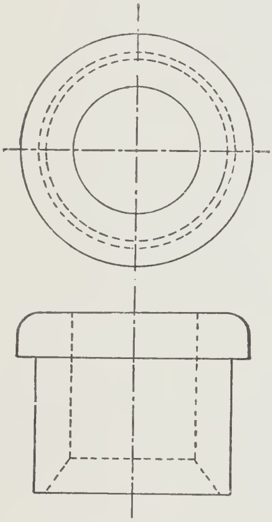


Fig. 8.

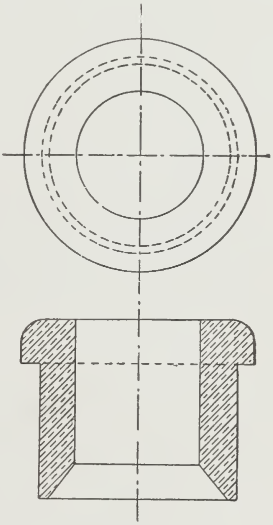


Fig. 9.

Consequently, most draftsmen place shade lines on all lines which represent lower and right-hand edges if these edges are sharp.

The contour lines of cylinders, cones and other rounded surfaces should not be shade lines, although some draftsmen shade them. If the cylinder is drawn in cross-section, however, the edge should be shaded, as the intersection of the plane and cylindrical surface is a sharp edge.

All views are shaded alike, and both are shaded as though they were elevations. The ray of light is supposed to come over the left shoulder of the draftsman, as he faces the paper, at such an angle that the projection of the ray of light on the drawing paper is in the direction of the arrow in Fig. 11.

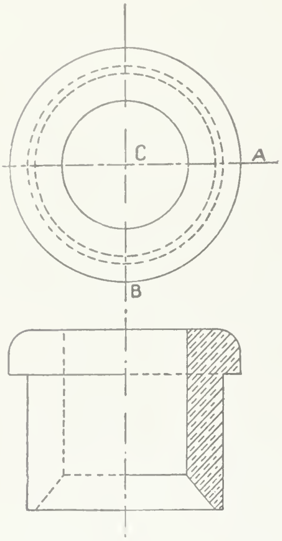


Fig. 10.

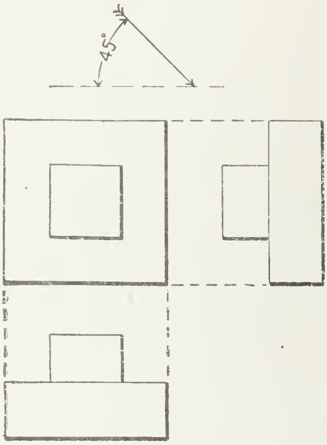


Fig. 11.

Figs. 11 to 18 show some of the most common shapes met with in drawings, and illustrate how the shade lines are placed on each. Fig. 11 is an elevation, plan, and side view of a rectangular prism with a smaller one resting on top of it. Fig. 12 is a plan and side view of a rectangular prism with a rectangular hole through it. It is to be noticed that the shade lines come on the upper and left-hand sides of the hole, since these lines are the

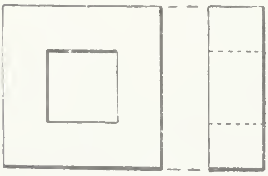


Fig. 12.

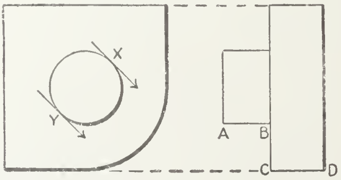
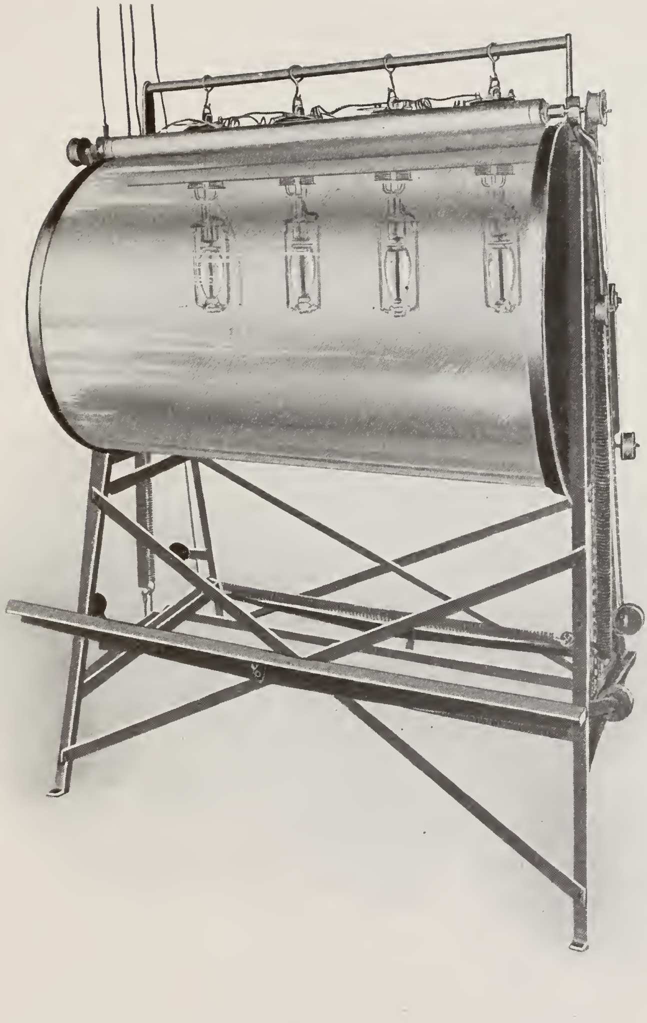


Fig. 13.



CIRCULAR BLUE PRINTING MACHINE, GENERAL ELECTRIC CO.

lower and right-hand edges of the material which surrounds the hole. Fig. 13 is a plan and side view of a rectangular prism with one corner rounded, and with a cylinder resting on it. Here the lines A B and C D are not shaded, since they are the contour lines of curved surfaces. In the plan view, the lower right-hand part of the circle, between X and Y, is shaded. To find these points X and Y, draw two lines tangent to the circle and making an angle of 45° with the T-square line as shown by the arrows; X and Y are the points where the arrows are tangent to the circle. Fig. 14 is a plan, elevation and cross-section of a cylinder. Here, in the plan, the larger circle is shaded on the lower side, and the circle which represents the hole is shaded on the upper side. The points where the shade begins are determined as explained for Fig. 13. The lines M N and O P are shaded since, as the cylinder is supposed to be cut open, these lines now represent sharp edges. Fig. 15 is a plan, elevation and side view of a hexagonal prism with its long diameter parallel to the bottom of the paper. Fig. 16 is the same except that here the short diameter of the prism is parallel to the bottom of the paper. Fig. 17 is a plan and end view of a rectangular block with a wide slot of the shape H I J K M L O N, cut through it lengthwise. The main point to which attention should be called here is that the line

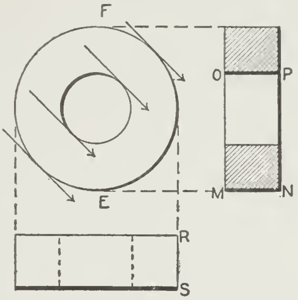


Fig. 14.

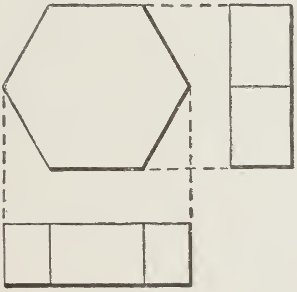


Fig. 15.

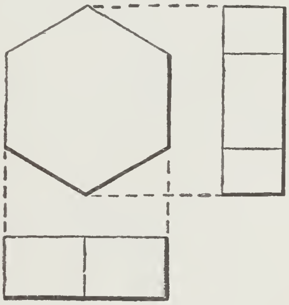


Fig. 16.

C D is shaded, although the slot might be so deep that the light might not strike in there because of the shadow of the projecting lip marked in the side view G H I. Fig. 18 is a cross-section and end view of a circular cylinder with a large hole extending part way through and a smaller hole extending the rest of the way. The small circle in the end view is shaded, although it is so far in that no light could strike it.

The student should study these figures carefully, and before leaving them should understand what each figure means and how the shade lines are determined, so that when he meets similar forms in machine drawings he may know where the shade lines should be placed.

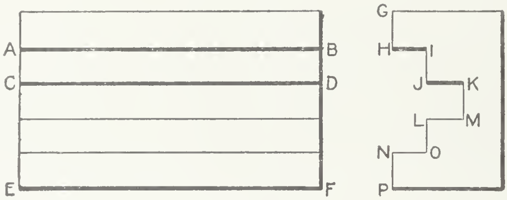


Fig. 17.

Dimensions. In giving the distance from one line to another, a dimension line is drawn between the two lines and arrowheads are placed at each end of the dimension line; the points of the arrowheads being *exactly on* the lines to which the measurement is being given. At a convenient place between the arrowheads a space is left in the dimension line for the figure.

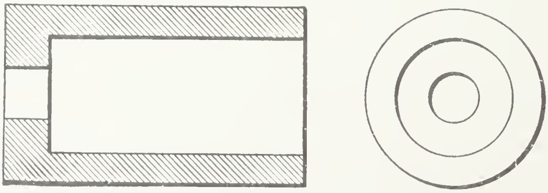


Fig. 18.

Fig 19 is a plan and elevation of a rectangular piece with a cylinder on the center of its upper surface, and with four holes through it; the holes being symmetrically located with respect to the center line of the piece; that is, the holes A and B are just as far above the center line E F as C and D are below it, and the holes B and D are just as far to the right of the center line G H as A and C are to the left of it. The fact that the cylinder is on the center of the rectangular piece is shown by the lines E F and G H being drawn through the center of the circle in the plan view, these lines being the center lines both of the cylinder and of the rectangular piece. If the cylinder were not on the center, two other center lines would be drawn through the circle and the dimensions given from each of these lines to the center lines of the rectangular piece, or to the edges of the piece. The fact that the holes are symmetrically located is shown by the dimensions being given between the center lines of the holes and no dimensions being given from these center lines to the center lines of the piece.

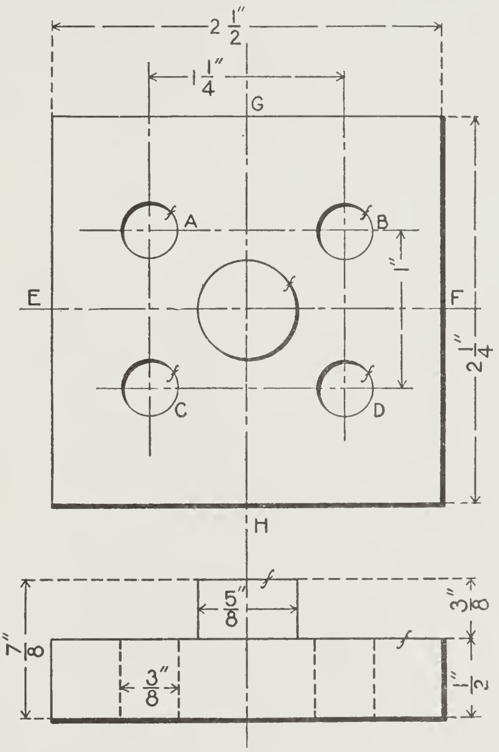


Fig. 19.

Fig. 20 shows how the piece might be dimensioned if the cylinder were not on the center of the rectangular piece and the holes were not symmetrically located with respect to the center of the rectangular piece.

If the centers of a set of holes are so located that a circle can be drawn through them, we always locate the holes by drawing a center line circle through their centers and giving the diameter of this circle, and if no other dimension than this is given for the location of the holes, it is understood that they are equally spaced around this circle. Fig. 21 illustrates this. If the holes are not equally spaced around the circle, they may be located as shown in Fig. 22.

Diameters of circles, and of arcs of circles which are greater than semicircles, should be given rather than radii, but if the arc of the circle is less than a semicircle, its radius should be given. Fig. 23 shows how this should be done when the radius is large enough to permit; but if the radius is small, or the dimensions would interfere with other parts of the drawing, it may be done as in Fig. 24.

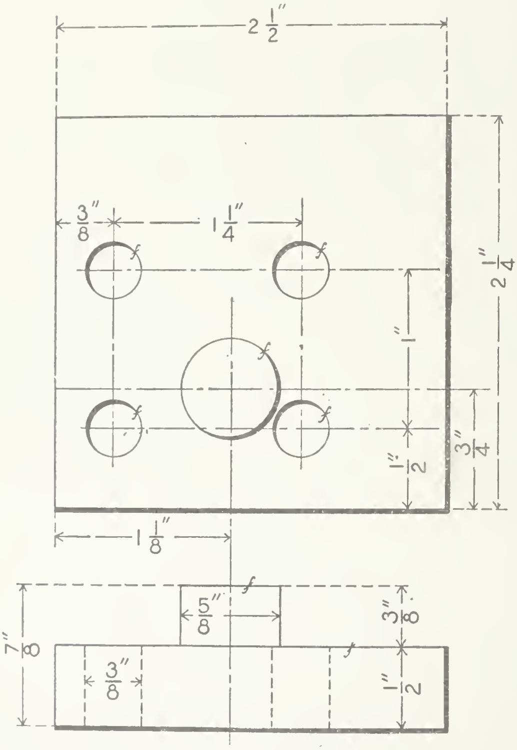
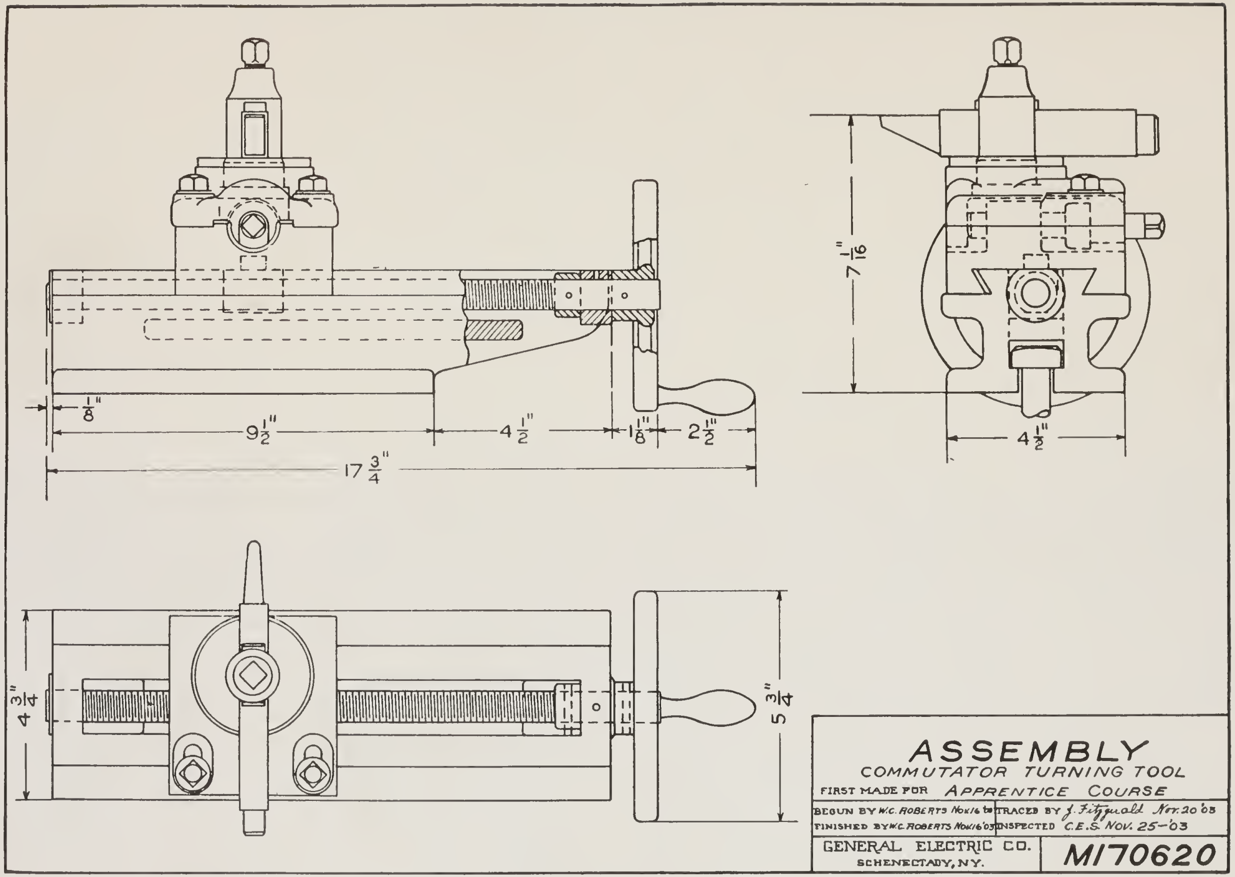


Fig. 20.

In Fig. 23 a small circle is drawn (free hand) around the center about which the arc is drawn, and a dimension line carried from the edge of this circle to the arc; an arrowhead being placed on the arc, and the figures placed in the dimension line in the usual way. In this case it is not necessary to put the letter R (abbreviation for radius) after the dimension. In the two methods shown in Fig. 24 the R should be placed after the figures.

In putting dimensions on a drawing, the figures should be placed to read from only two sides of the paper, usually from the bottom for dimension lines which are horizontal, as the $2\frac{1}{2}''$ dimen-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9 1/2"
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ASSEMBLY
COMMUTATOR TURNING TOOL
FIRST MADE FOR APPRENTICE COURSE
BEGUN BY W.C. ROBERTS Nov.16 '03
TRACED BY J. Fitzgerald Nov. 20 '03
FINISHED BY W.C. ROBERTS Nov.16 '03
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EXAMPLE OF SHOP DRAWING MADE IN DRAFTING ROOM OF THE GENERAL ELECTRIC COMPANY.

sion in Fig. 20, and from the right hand for dimension lines which are vertical, as the $2\frac{1}{4}''$ dimension in the same figure. Dimensions should never be placed on center lines if it can be avoided.

If a dimension is given in one view of an object, the same dimension need not be repeated in the other views.

In putting on fractions it is better to have the line which divides the numerator and denominator of the fraction extend in the same direction as the dimension line rather than at an angle; that is, the fraction would appear thus, $\frac{1}{2}$, rather than $^1/_2$.

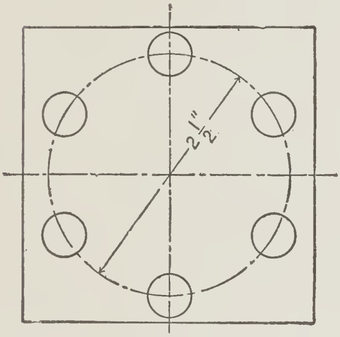


Fig. 21.

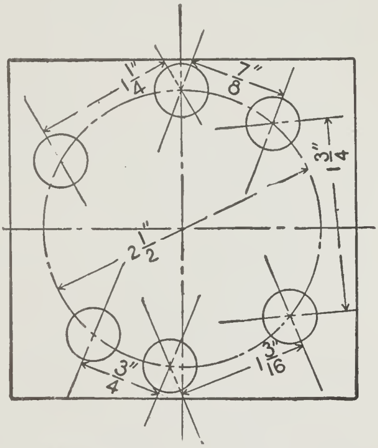


Fig. 22.

Finished Surfaces. Since a working drawing is not only to show the shape and size of the object which it represents, but is to describe it completely, it is essential that there should be some means of distinguishing between the surfaces of the object which are to be left rough as they come from the forge or from the foundry, and the surfaces which are to be finished. Any surface

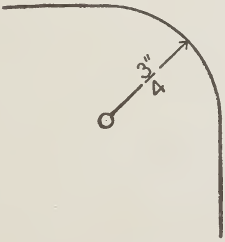


Fig. 23.

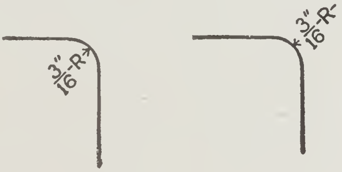


Fig. 24.

which has been smoothed off in a lathe, planer, or any machine tool, or has been filed or scraped, is called a finished surface, and is indicated on a drawing by a letter placed on the *edge view* of this surface. In Fig. 19 the whole of the cylindrical part is finished, also the top of the rectangular part. The holes are also finished. The student should notice carefully how the finish marks are used to indicate this fact.

If a piece is to be finished all over, it should be marked "*f* all over." If it is desired to specify what kind of finish is to be put on a surface, that is, whether rough turned, smooth turned, filed, scraped, etc., a note may be made to that effect.

On some conspicuous place on the drawing the words "*f* means finish" should be printed, so that there may be no misunderstanding on the part of a person examining the drawing.

Material. The material of which the piece is to be made should be indicated plainly. If any part of the drawing is a cross-section, the crosshatching might show the material, provided that in some conspicuous part of the drawing a sample of the crosshatching is shown and the material which it represents is stated. It is better, however, to mark on or near each piece, in plain letters, the material of which it is to be made.

Conventional Methods. In drafting, as in many other kinds of work, all unnecessary labor should be avoided. The drawing should give the instructions clearly, but need not be so elaborate as to require unnecessary time in the execution. It frequently happens that if an exact drawing were made representing every line of the object, a great deal of time would be required. Accordingly, if there is any way of representing an object by a few lines only, without sacrificing clearness, it should be used. There are many details such as screw threads, springs, bolts, etc., which occur so frequently on nearly all drawings that easy methods of representation have been universally adopted. In some cases, these ways of representing objects are approximations of the exact drawings and in other cases they are not. Such methods are called conventional methods, or conventions. For some details there is more than one convention, but the same convention should not be used for two different things, nor should several objects of the same kind on the same drawing be represented conventionally by two different methods.

Screw Threads. The exact drawing of screw threads is a difficult part of mechanical drawing, but enough attention will be given it here to enable the student to make a simple working drawing which includes threads. The common conventional way of drawing a thread is shown in Figs. 25, 26 and 27, Fig. 25 being a single right-hand thread, and Fig. 26 a double right-hand thread.

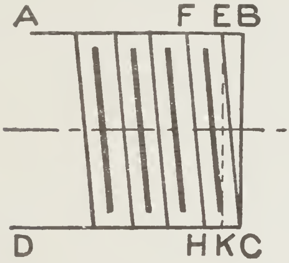


Fig. 25.

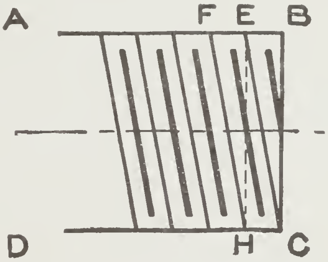


Fig. 26.

First, the plain cylindrical piece ABCD is drawn as if there were no threads upon it. The thread is then indicated by the lines EC, FH, etc., with the shorter and heavier lines between. The lighter lines are all parallel and the same distance apart. This distance is not necessarily the same as the actual pitch of the thread on the screw itself, but is usually $\frac{3}{32}$ inch or $\frac{1}{8}$ inch on drawings of common sizes of bolts. The heavy lines are parallel to the lighter ones and midway between them. The angle which the lines make with the center line of the screw depends on the distance apart of the lines, the slant being such that for a single thread, Fig. 25, a line perpendicular to the center line through one end of one of the lighter lines, as E, will strike the opposite contour of the screw DC, at a point K, which is half way between C and H. To draw the lines so that they will have this slant, choose the distance FE, which is to represent the pitch, and space it off on either of the contour lines, as AB, starting anywhere (in the figure the starting point E is one-half of the distance FE away from B). Next, draw EK per-

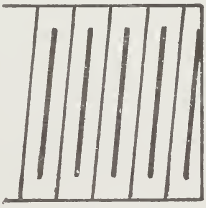


Fig. 27.

pendicular to the center line, thus finding point K, and start the spacing on the contour CD at a distance either side of K equal to $\frac{1}{2}$ FE.

The lighter lines can thus be drawn and the heavy lines put in parallel and half way between. The heavy lines should be a little shorter than the others, and, for the sake of neatness, all should be of the same length. The double thread, Fig. 26, is drawn similarly to the single thread, except that the slant is such that the perpendicular through E will pass through H.

After a little practice the student can draw the threads in this way with his triangles, getting the proper slant and the angle spacing by eye, without the necessity of measuring; he should practice with this end in view, for threads occur so frequently on drawings that the draftsman must be able to draw them rapidly.

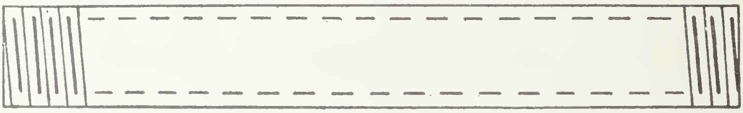


Fig. 28.

Notice should be taken of the fact that if the page be held so that the center lines of the screws in Figs. 25 and 26 are vertical, the lines which represent the threads slant downward from right to left. If the thread is left-handed the lines slant from left to right, as in Fig. 27.

In case of a long screw a few threads may be represented at each end and dotted lines carried the rest of the way, as in Fig. 28.

On any drawing of a screw which is intended for a working drawing, the pitch of the thread, or more commonly the number of threads for an inch of length should be specified; when the thread be standard, it is not always necessary to specify. Even in that case it is well to state the fact that the thread is standard. In giving the number of threads per inch it may be abbreviated, thus 12 THDS, or 12 TH. Examples of this will occur in what follows.

Threaded Holes and Invisible Threads. Fig. 29 shows a piece with a threaded hole in it. The hole is represented in the side view by the four parallel dotted lines. The distance apart of the two outer lines is equal to the diameter of the piece which is to be screwed into the hole. The inner lines are at a distance from the outer approximately equal to the depth of the thread. In the plan view of the piece (that is, the view looking at the end of the hole) the hole is shown by a full circle of a diameter equal to the distance apart of the inner dotted lines of the other view, and around this full circle a dotted circle whose diameter is equal to the diameter of the bolt; or, in other words, equal to the distance apart of the outer dotted straight lines. The dimension might be given on either view. In the figure it is given on the side view. In giving the diameter of a threaded hole, the diameter of the piece which is to be screwed into the hole is always given. In Fig. 29, the number of threads per inch is not stated, and in this case it would be understood to be standard. Fig. 30 shows another way of dimensioning a tapped hole, which is satisfactory and convenient.

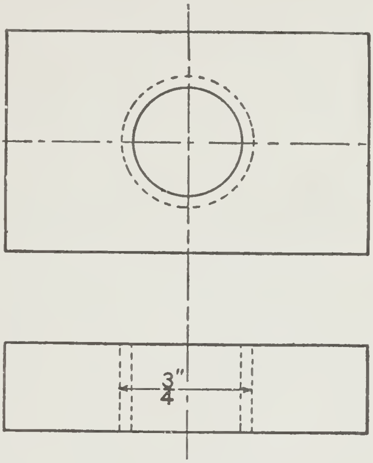


Fig. 29.

Fig. 31 shows the side view of the same piece as Fig. 29, with a part of the screw inserted in the hole. Attention is called to the fact that where the screw is hidden as it goes through the hole the thread is not shown, the parallel dotted lines representing the thread in the same manner as when the hole had no screw in it. Another point to be noted is that the cross-hatching of the broken ends for wrought iron is similar to the representation of the screw thread and one must learn to judge when the light and heavy lines mean a thread and when they are cross-hatching.

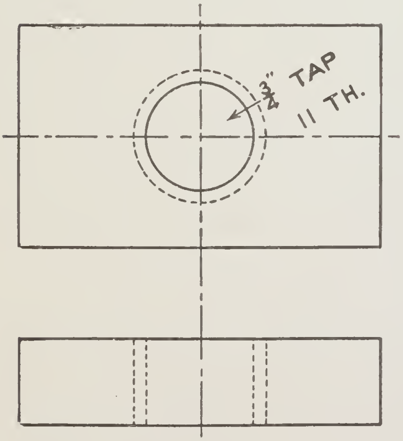


Fig. 30.

Threads in Sectional Pieces. Figs. 32, 33 and 34 illustrate the common method of representing threads when they occur on pieces which are drawn in cross-section. Fig. 32 is the same piece as Fig. 29, shown cross-sectioned. The front half of the piece is supposed to be removed and we are looking at the back half. Now the thread on the back side of a screw slants the opposite way from what it does on the front side, and of course the same is true of the thread in a tapped hole. Consequently, since it is the back side of the hole which is seen, the slant of the lines which represent the thread is opposite to the direction they would have were we looking at the front side of the screw which goes into the hole. We have just learned that for a right-handed thread on a screw the lines slant downward from right to left, and therefore for a right-handed thread seen on the back side of a tapped hole, the lines will slant downward from left to right. In other words, for a right-handed thread in a hole which comes in a cross-section, the lines slant the same as they would on the front of a left-hand thread on a bolt; and for a left-hand thread in a sectioned hole, he slant is the same as for a right-hand thread on a bolt.

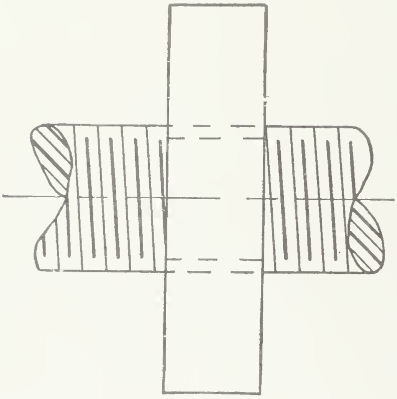


Fig. 31.

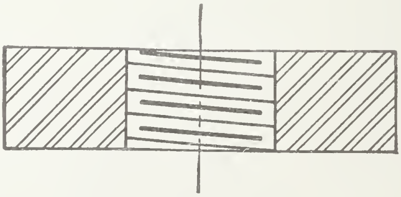


Fig. 32.

Fig. 33 is a piece which has a smooth hole through it and a thread on the outside. Here the entire thread is invisible, except at the contour of the cylinder, and must be indicated by the notches. These are drawn by spacing off the distance which is used for the pitch and from the points thus found drawing lines with the triangle which make an angle of 60 degrees with the axis of the cylinder. For a single thread the notches on one side have their outer points opposite the inner points of the notches on the other side. For a double thread the notches are directly opposite each other.

Fig. 34 shows two ways of quarter-sectioning a threaded piece, the only difference being that on one the contour of the sectional part is drawn a straight line, while on the other the contour is notched. Either one may be used. The straight contour can, of course, be drawn much more quickly and in places where there is no danger of sacrificing clearness it should be used for that reason. If the drawing is somewhat complicated, so that without the notches it might not be quite clear that the piece was threaded, the notches should be used.

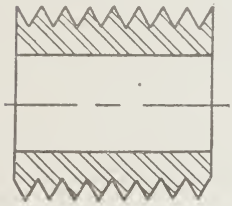


Fig. 33.

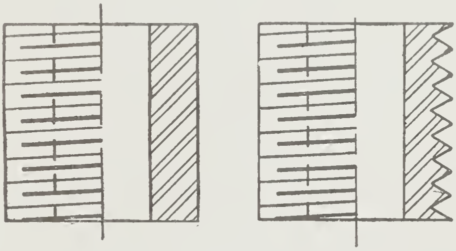


Fig. 34.

As has already been suggested, the student will doubtless find many other customs in the matter of drawing threads which are quite as good as the above. These have been given as ones which are common, and easily drawn. As a matter of convenience the following tables are given, which show the number of threads per inch on some of the most common sizes of bolts, according to the standard adopted by the United States Government, and the Whitworth or English standard.

UNITED STATES STANDARD SCREW THREADS.

Diameter of Bolt.	Threads per Inch.	Diameter of Bolt.	Threads per Inch.	Diameter of Bolt.	Threads per Inch.
$\frac{1}{4}$	20	$\frac{5}{8}$	11	$1\frac{3}{8}$	6
$\frac{5}{16}$	18	$\frac{3}{4}$	10	$1\frac{1}{2}$	6
$\frac{3}{8}$	16	$\frac{7}{8}$	9	$1\frac{5}{8}$	$5\frac{1}{2}$
$\frac{7}{16}$	14	1	8	$1\frac{3}{4}$	5
$\frac{1}{2}$	13	$1\frac{1}{8}$	7	$1\frac{7}{8}$	5
$\frac{9}{16}$	12	$1\frac{1}{4}$	7	2	$4\frac{1}{2}$

WHITWORTH STANDARD SCREW THREADS.

Diameter of Bolt.	Threads per Inch.	Diameter of Bolt.	Threads per Inch.	Diameter of Bolt.	Threads per Inch.
$\frac{1}{4}$	20	$\frac{5}{8}$	11	$1\frac{3}{8}$	6
$\frac{5}{16}$	18	$\frac{3}{4}$	10	$1\frac{1}{2}$	6
$\frac{3}{8}$	16	$\frac{7}{8}$	9	$1\frac{5}{8}$	5
$\frac{7}{16}$	14	1	8	$1\frac{3}{4}$	5
$\frac{1}{2}$	12	$1\frac{1}{8}$	7	$1\frac{7}{8}$	$4\frac{1}{2}$
$\frac{9}{16}$	12	$1\frac{1}{4}$	7	2	$4\frac{1}{2}$

Bolts and Nuts. Among the most common pieces of machinery are bolts and nuts, and the draftsman has frequent occasion to draw them. Fig. 35 is a conventional drawing of a $\frac{1}{2}$-inch bolt with a hexagonal head and nut. This figure shows the dimensions necessary to be given in order that a workman shall be able to make the bolt from the drawing.

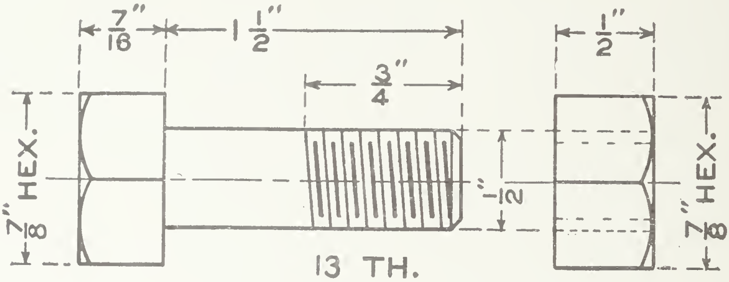


Fig. 35.

Beginning with the head of the bolt, let us study the various parts of this drawing in detail. The head is a hexagonal prism the end of which has been chamfered. We might expect that two views of the head would be necessary to completely define its shape, but the letters HEX printed after the dimension for the diameter of the head indicates that it is hexagonal. In like manner, if the head were square the letters SQ would be placed after the dimension. If two views were drawn, they would appear as in Fig. 36 The head is drawn as if the tool which cut the chamfer, cut off just the corners of the top, so as to make the top a circle tangent to the sides of the hexagon at B, E, G, etc.; the parts BCE, EFG, etc.. being portions of a cone or sphere according to the manner of chamfering. The curves ABC and CEF are tangent to the line

GK in the side view, and the lines AJ, CD, and FH are all equal in length. The curves ABC and CEF are properly the lines of intersection of a cone or sphere, as the case may be, with the hexagonal prism, but a convenient and sufficiently accurate way of drawing them is by arcs of circles with the center on the line HJ, half way between HD and DJ, as indicated. The chamfer is represented in the end view by a circle inscribed in the hexagon.

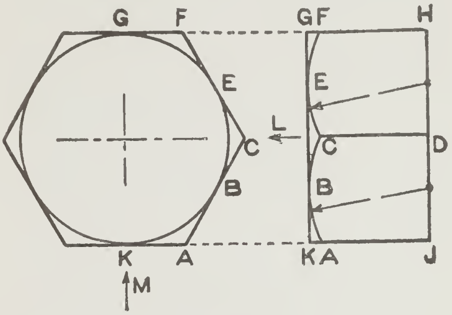


Fig. 36.

It will be noticed that in Fig. 35 that view of the head is given which shows two faces of the prism, so that the shortest dimension of the hexagon is given. That is, the view is taken in the direction of the arrow L, Fig. 36, instead of in the direction of the arrow M.

This rule should be followed on all detail drawings. The case when it is desirable to give a view in the direction of the arrow M will be discussed later.

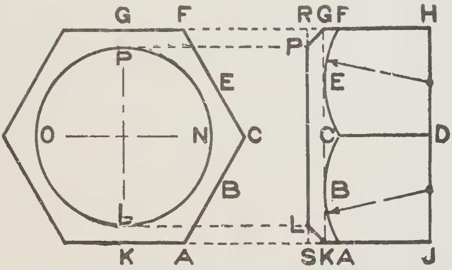


Fig. 37.

Sometimes it is desired to represent the chamfer as being greater, as in Fig. 37. To do this, draw the end view of the hexagon and inside it, of a diameter as much smaller as desired, draw the circle PNLO. Next draw the side view of the hexagonal prism as if there were no chamfer, as shown at the right (partly in dotted lines). Then from P and L, the upper and lower points of the circle in the other view, draw lines perpendicular to RS and meeting it at P and L. From P and L draw PG and LK, either at an angle of 45 degrees or 30 degrees (the latter being preferable) with RS, and meeting the lines RH and SJ at G and K. Join G and K by a construction line GK (shown dotted but not to be left in the finished drawing). Draw the arcs ABC and CEF with center on line HJ as shown, and tangent to GK.

Fig. 38 shows the proper way of representing the head which was shown in Fig. 36 if the view is taken in the direction of the arrow M. The dotted construction lines show how the widths are obtained. Fig. 39 shows the corresponding view of the head shown in Fig. 37. The radius with which the middle arc representing the chamfer is drawn in these two figures is apparent. Various rules are given for finding a radius for small arcs, but they can be found near enough by trial, after a little experience. The line TV, Fig. 39, to which the tops of the arcs are tangent, is drawn parallel to RS and at a distance from it equal to RG in Fig. 37.

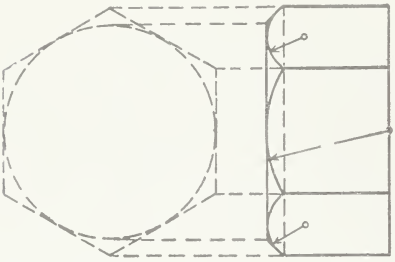


Fig. 38.

The nut, Fig. 35, is drawn in exactly the same way as the head, so that what has just been said will apply equally well to the nut. The hole in the nut is indicated by the four parallel dotted lines as explained for Fig. 29.

The shank of the bolt is represented with the thread upon it as explained above for conventional threads. The point is drawn chamfered a little in the figure, so that it appears as the frustum of a cone. Bolts often have round points, in which case they would be shown as in Fig. 40. The lines which represent the thread should not cross the line which is drawn square across the bolt to indicate where the chamfer or the rounding of the point begins.

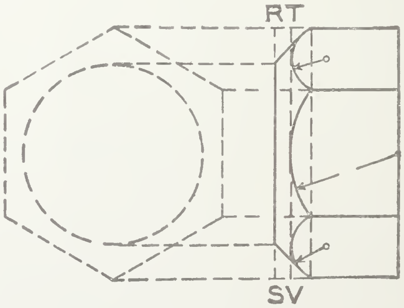


Fig. 39

Fig. 41 shows three views of a square head or nut with chamfer corresponding to that on the hexagonal head in Fig. 36: and Fig. 42 shows the square head or nut chamfered to correspond to

Fig. 37. Referring first to Fig. 41 the arc on the side view which shows the short diameter of the nut is drawn with a radius A, equal to two and one-quarter times the diameter of the bolt on which the head or nut belongs. The arcs on the other side view are drawn with a radius B, equal to one-half of A. The lines EF are drawn from points E tangent to the arcs, and it will be found that the points of tangency will come almost at the points where the arcs cut the lines CD. Points E are found by projecting from the plan view as indicated by the dotted lines. In Fig. 42, the construction is similar. The points N are first found by projecting from the top and bottom of the circle in the plan view; then the lines NL are drawn making angles of 30 degrees with line NN. (The proportions for the radii which are given, hold good only when the angle of 30 degrees is used). Next draw the construction line LL and draw the arc tangent to it with a radius A equal to two and one-quarter times the diameter of the bolt, the same as in Fig. 41. To draw the chamfer in the other side view, draw the construction line parallel to and at a distance from CC equal to the distance LL from NN and draw the arcs

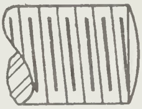


Fig. 40.

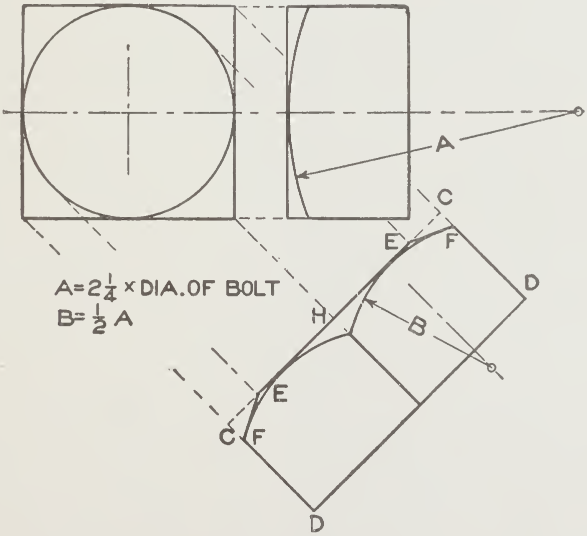


Fig. 41.

tangent to this line with radius B equal to one-half of A. The lines EF are then drawn as explained for Fig. 41.

Referring again to Fig. 35, the dimension which shows the length of the bolt under the head should be given to the extreme point of the bolt, as should also the dimension which shows how much of the bolt is threaded.

Most of the bolts in common use are made of standard sizes, that is, for a certain diameter of bolt there is a corresponding standard diameter and thickness for the head and the nut, and a standard number of threads per inch, so that if the bolt which the draftsman wishes to use has these standard dimensions they may be omitted from the drawing and a note made that the bolt is standard. Then the only dimensions necessary to be given are the diameter, the length under the head, and the length of the threaded part.

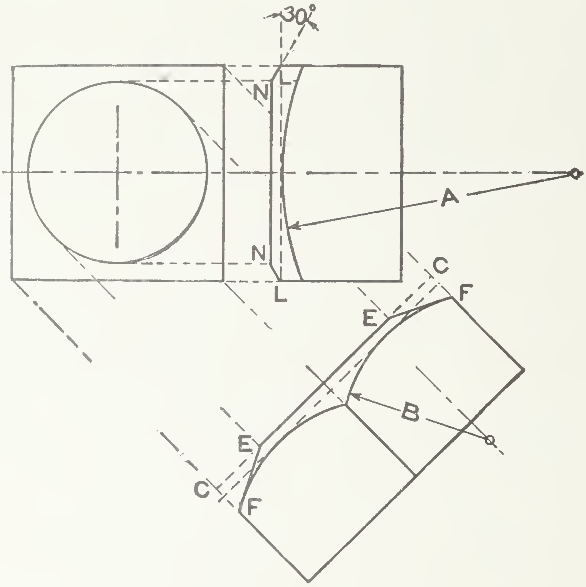


Fig. 42.

The following tables give the United States standard sizes of square and hexagonal heads and nuts for bolts. The columns headed "Width of Nut" and "Width of Head" give the shortest dimension of the square or hexagon, that is, the diameter of the inscribed circle. The standard number of threads per inch can be found from the table already given.

SQUARE BOLT HEADS. U. S. STANDARD (Franklin Institute).

Dia. of Bolt.	Width of Head.	Thickness of Head.	Dia. of Bolt.	Width of Head.	Thickness af Head.
$\frac{1}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	1	$1\frac{5}{8}$	$\frac{13}{16}$
$\frac{5}{16}$	$\frac{19}{32}$	$\frac{19}{64}$	$1\frac{1}{8}$	$1\frac{13}{16}$	$\frac{29}{64}$
$\frac{3}{8}$	$\frac{11}{16}$	$\frac{11}{32}$	$1\frac{1}{4}$	2	1
$\frac{7}{16}$	$\frac{25}{32}$	$\frac{25}{64}$	$1\frac{3}{8}$	$2\frac{3}{16}$	$1\frac{3}{32}$
$\frac{1}{2}$	$\frac{7}{8}$	$\frac{7}{16}$	$1\frac{1}{2}$	$2\frac{3}{8}$	$1\frac{3}{16}$
$\frac{9}{16}$	$\frac{31}{32}$	$\frac{31}{64}$	$1\frac{5}{8}$	$2\frac{9}{16}$	$1\frac{9}{32}$
$\frac{5}{8}$	$1\frac{1}{16}$	$\frac{17}{32}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$1\frac{3}{8}$
$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{5}{8}$	$1\frac{7}{8}$	$2\frac{15}{16}$	$1\frac{15}{32}$
$\frac{7}{8}$	$1\frac{7}{16}$	$\frac{23}{32}$	2	$3\frac{1}{8}$	$1\frac{9}{16}$

HEXAGON BOLT HEADS. U. S. STANDARD (Franklin Institute).

Dia. of Bolt.	Width of Head.	Thickness of Head.	Dia. of Bolt.	Width of Head.	Thickness of Head.
$\frac{1}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	1	$1\frac{5}{8}$	$\frac{13}{16}$
$\frac{5}{16}$	$\frac{19}{32}$	$\frac{19}{64}$	$1\frac{1}{8}$	$1\frac{13}{16}$	$\frac{29}{64}$
$\frac{3}{8}$	$\frac{11}{16}$	$\frac{11}{32}$	$1\frac{1}{4}$	2	1
$\frac{7}{16}$	$\frac{25}{32}$	$\frac{25}{64}$	$1\frac{3}{8}$	$2\frac{3}{16}$	$1\frac{3}{32}$
$\frac{1}{2}$	$\frac{7}{8}$	$\frac{7}{16}$	$1\frac{1}{2}$	$2\frac{3}{8}$	$1\frac{3}{16}$
$\frac{9}{16}$	$\frac{31}{32}$	$\frac{31}{64}$	$1\frac{5}{8}$	$2\frac{9}{16}$	$1\frac{9}{32}$
$\frac{5}{8}$	$1\frac{1}{16}$	$\frac{17}{32}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$1\frac{3}{8}$
$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{5}{8}$	$1\frac{7}{8}$	$2\frac{15}{16}$	$1\frac{15}{32}$
$\frac{7}{8}$	$1\frac{7}{16}$	$\frac{23}{32}$	2	$3\frac{1}{8}$	$1\frac{9}{16}$

SQUARE AND HEXAGON NUTS. U. S. STANDARD (Franklin Institute).

Dia. of Bolt.	Width of of Nut.	Thickness of Nut.	Dia. of Bolt.	Width of Nut.	Thickness of Nut.
$\frac{1}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	1	$1\frac{5}{8}$	1
$\frac{5}{16}$	$\frac{19}{32}$	$\frac{5}{16}$	$1\frac{1}{8}$	$1\frac{13}{16}$	$1\frac{1}{8}$
$\frac{3}{8}$	$\frac{11}{16}$	$\frac{3}{8}$	$1\frac{1}{4}$	2	$1\frac{1}{4}$
$\frac{7}{16}$	$\frac{25}{32}$	$\frac{7}{16}$	$1\frac{3}{8}$	$2\frac{3}{16}$	$1\frac{3}{8}$
$\frac{1}{2}$	$\frac{7}{8}$	$\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{3}{8}$	$1\frac{1}{2}$
$\frac{9}{16}$	$\frac{31}{32}$	$\frac{9}{16}$	$1\frac{5}{8}$	$2\frac{9}{16}$	$1\frac{5}{8}$
$\frac{5}{8}$	$1\frac{1}{16}$	$\frac{5}{8}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$1\frac{3}{4}$
$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$1\frac{7}{8}$	$2\frac{15}{16}$	$1\frac{7}{8}$
$\frac{7}{8}$	$1\frac{7}{16}$	$\frac{7}{8}$	2	$3\frac{1}{8}$	2

Pipes and Pipe Threads. The various kinds of pipe in common use are made to standard sizes, and as the draftsman very often comes in contact with piping we will consider it briefly. The kinds most often used are wrought-iron or steel pipe, brass pipe made to the size of wrought-iron pipe, and cast-iron pipe. The cast-iron pipe is made of different weights and form, according to the purpose for which it is to be used. Standard weight iron pipe is rated by its nominal inside diameter, although the actual diameter does not in most cases quite agree with the nominal diameter. A $\frac{1}{4}$-inch pipe is a pipe, the hole in which is supposed to be $\frac{1}{4}$ inch in diameter, but if carefully measured it will be found to be a few hundredths of an inch larger.

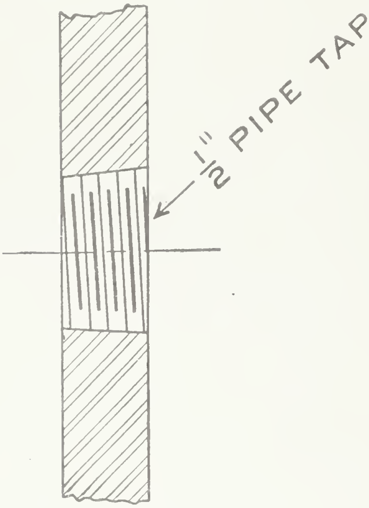


Fig. 43.

The threads on pipes and pipe fittings are also made to standard; taps and dies are made for various sizes of pipe. These taps and dies are spoken of or described by stating the size of the pipe for which they are intended For example, a $\frac{1}{4}$-inch pipe-tap is a tap of the proper size, shape, and number of threads per inch to cut the thread in a hole to receive a $\frac{1}{4}$-inch pipe. Threaded holes are made tapering for pipes, the standard taper being $\frac{3}{4}$ inch per foot, that is, the diameter of the hole decreases at the rate of $\frac{3}{4}$ inch per foot. In representing a hole which is threaded with a pipe tap, the hole is drawn of a diameter at its larger end about equal to the outside diameter of the pipe which is to be screwed into it, and is drawn tapering. It is well to make the taper considerably greater than the actual taper, so that the person looking at the drawing may see at a glance that the hole is for a pipe.

The thread is indicated in one of the conventional ways previously explained, but the number of threads per inch and the diameter of the hole need not be given; instead, a note is made that the hole is tapped for a certain size pipe. Fig. 43 illustrates this.

The following tables of standards for wrought-iron pipe may be found convenient:

STANDARD SIZES OF WROUGHT IRON PIPE.

Nominal Size.	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2
Actual Inside Diameter.	.27	.364	.494	.623	.824	1.048	1.38	1.610	2.067
Outside Diameter.	.405	.54	.675	.84	1.05	1.315	1.66	1.90	2.375
Nominal Size.	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$	5	6	7	8
Actual Inside Diameter.	2.468	3.067	3.548	4.026	4.508	5.045	6.065	7.023	7.982
Outside Diameter.	2.875	3.50	4.00	4.50	5.00	5.563	6.625	7.625	8.625

STANDARD THREADS FOR WROUGHT IRON PIPE.

Nominal Size of Pipe.	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2
Threads per Inch.	27	18	18	14	14	$11\frac{1}{2}$	$11\frac{1}{2}$	$11\frac{1}{2}$	$11\frac{1}{2}$
Nominal Size of Pipe.	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$	5	6	7	8
Threads per Inch.	8	8	8	8	8	8	8	8	8

Scale Drawings. When the object which is to be drawn is not so large but that it can be easily actual size, or full size as it is called, on a sheet of paper which is of convenient dimensions, it is well, usually, to draw the piece full size. In most cases, however, the machine, or the building, or whatever is to be drawn is so large that it would be impossible to draw it full size. Then the drawing is made to some reduced scale, that is, all the dimensions are drawn smaller than the actual dimensions of the object itself; all dimensions being reduced in the same proportion. For example, if a piece is to be drawn half size, the distance from one point to another on the drawing would be one-half what it is on the piece itself; if the drawing is one-fourth size, the distance on the drawing would be one-fourth what it is on the piece itself, and so on. In dimensioning such a drawing the dimension which is written on the drawing is the *actual dimension* of the piece, and not the distance which is measured on the drawing. This fact must be very clearly understood by the student.

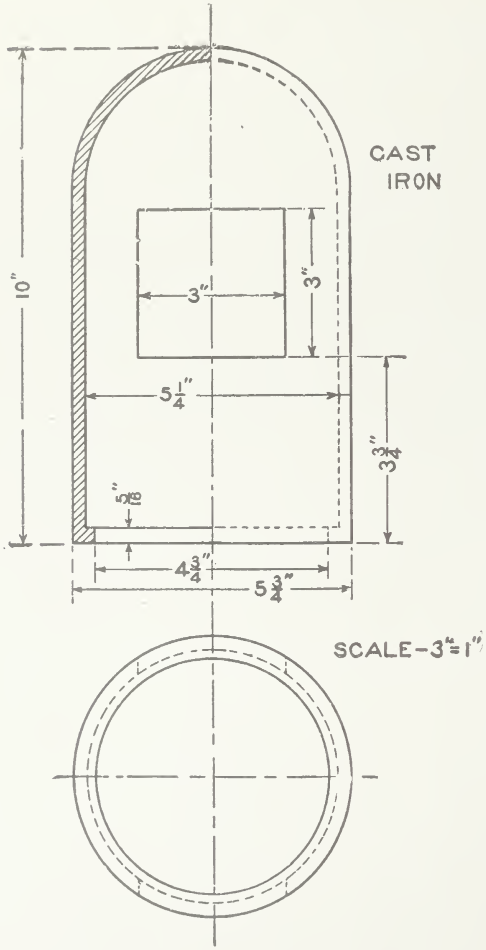


Fig. 44.

The common method of reducing all the dimensions in the same proportion is to choose a certain distance and let that distance represent one foot, this distance is then divided into twelve parts and each one of these parts represents an inch; then if half and quarter inches are required these twelfths are subdivided into halves, quarters, etc., until the subdivisions become so small that they cannot be used. We now have a scale which represents the common foot rule with its subdivisions into inches and fractions; but our new foot is smaller than the ordinary distance which we call a foot, and of course its subdivisions are proportionately smaller. When we make a measurement on the drawing we make it with our reduced foot rule and when we make a measurement on the machine itself we make it with the common foot rule.

Draftsmen's scales can be bought which have different distances thus divided, so that if the draftsman wishes to draw a piece one-fourth size he looks over his scale until he finds a distance of three inches (which is of course one-fourth of a foot) divided as explained above, and he uses this to measure with on his drawing. His drawing would then be made to a scale of three inches to the foot. In the same way, if he wishes to make his drawing one-twelfth size he finds on his scale one inch divided into twelfths and fractions of twelfths and uses this as his standard of measurement; if he wishes to make his drawing one forty-eighth size he uses a quarter inch with its subdivisions. Sometimes if the piece to be drawn is very small, the drawing is made at an enlarged scale, such as twice size, three times size, etc.

The mistake of choosing the wrong distance to use on a scale is often made. For example, if he wishes to draw a piece $\frac{1}{4}$ size, he will look over his scale for a place marked $\frac{1}{4}$, and use this for his standard for $\frac{1}{4}$ size, which is wrong. The figure on the scale indicates the distance which is divided up to represent one foot, so that the part of the scale which has $\frac{1}{4}$ marked on it means that $\frac{1}{4}$ of an inch is divided up into twelfths, or in other words, if a drawing is made according to that scale it will be $\frac{1}{48}$th size.

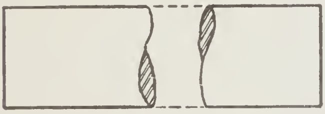


Fig. 45.

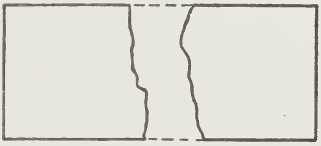


Fig. 46.

Every drawing which is made at any other scale than full size should have the scale marked on it plainly.

Fig. 44 shows a piece drawn to a scale of 3 inches per foot, that is, $\frac{1}{4}$ size.

Long Pieces. A piece which is very long in proportion to its diameter or width is often difficult to draw complete, especially if there is much detail to any part of it, for if the scale is made so small that the length will go on a sheet of paper of convenient size the small part is so reduced that it is very small. If such a piece is plain the whole or part of its length, a portion of the plain part may be broken out on the drawing thus shortening the drawing of the piece so that a larger scale may be used. Of course in giving the dimension for the length, the *actual* length must be given. Fig. 45 shows a round piece thus broken out and Fig. 46 a rectangular piece.

It is a good idea to connect by dotted lines the two parts thus broken, although this is not essential.

"Turning Up" a Section. Sometimes a second view of a piece may be avoided by drawing on the first view a partial view showing the shape of the cross-section at the place where drawn. This partial view or "turned-up section" may be drawn in either full or dotted lines and should be cross hatched. If it comes where there are other lines of the original view, as is usually the case, the original lines would be drawn in regardless of the fact that they conflicted with the auxiliary view. Fig. 47 is an illustration.

In General. A working drawing will usually belong to one of the three following classes: *first*, a design of an entirely new machine; *second*, a drawing of a machine which is already built; *third*, a drawing of a new part to fit a machine which is already built, or a drawing of an old machine remodelled.

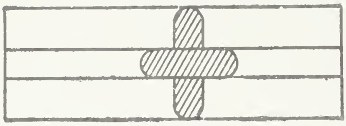


Fig. 47.

In order to design a new machine, or a part of a machine, the draftsman must understand the principles of drawing and must also have a clear understanding of the work which the machine is to perform, and must know something of the principles of machine design and the strength of materials. A study of the general manner of proceeding with the drawing, when these things are known, will come later in this course. The draftsman may make a drawing of a machine which is already built, even if he has no understanding of the working of the machine, although under such conditions he works at a great disadvantage. In any case, he must know the principles of drawing.

If the student has learned thoroughly what has preceded in this course, he should be ready to take up the drawing from a machine, and we will now consider the general system to be followed in making such a drawing.

Sketches. In most cases it would be inconvenient for the draftsman to take his drawing board and instruments to the machine and make his careful drawing with the machine close at hand; preferably, he makes what are called sketches, carries his sketches to the drafting room and makes his drawing from them. In making the sketches, each piece of the machine should be taken separately and a complete working drawing made of it, the only difference between these sketches and the finished drawing being that they are made largely without the use of instruments, triangles, etc.; that is, they are made free hand. The experienced draftsman will draw some lines and circles free hand, and use his instruments on some, according as he may think that he will save time by doing the one or the other; but it is well for the beginner to gain practice by making the sketches, almost wholly free hand, except large circles, and long lines.

The sketches should be neat and perfectly clear, so that if they are laid aside for a long time they can be clearly understood without depending at all upon memory. There is a strong tendency for the beginner to make his sketches hurriedly, thinking that when he comes to finish his drawing he can supply the details from memory. This is a bad plan and will lead to many mistakes The sketches must be so clear and complete that anyone can read them who has never seen the machine. No attempt need be made to draw them to scale, but **all dimensions,** carefully measured from the machine, should be placed on the sketch.

After every piece of the machine has been thus sketched separately, it is well to make a rough, general sketch of the whole machine, to show how the various pieces fit together, a few of the most important over-all dimensions, distances between centers, etc.

All sketching should be done as rapidly as possible without sacrificing clearness. Before starting to sketch a piece, the draftsman must decide what views are necessary to describe the piece clearly. All sketches should be made large to avoid confusion.

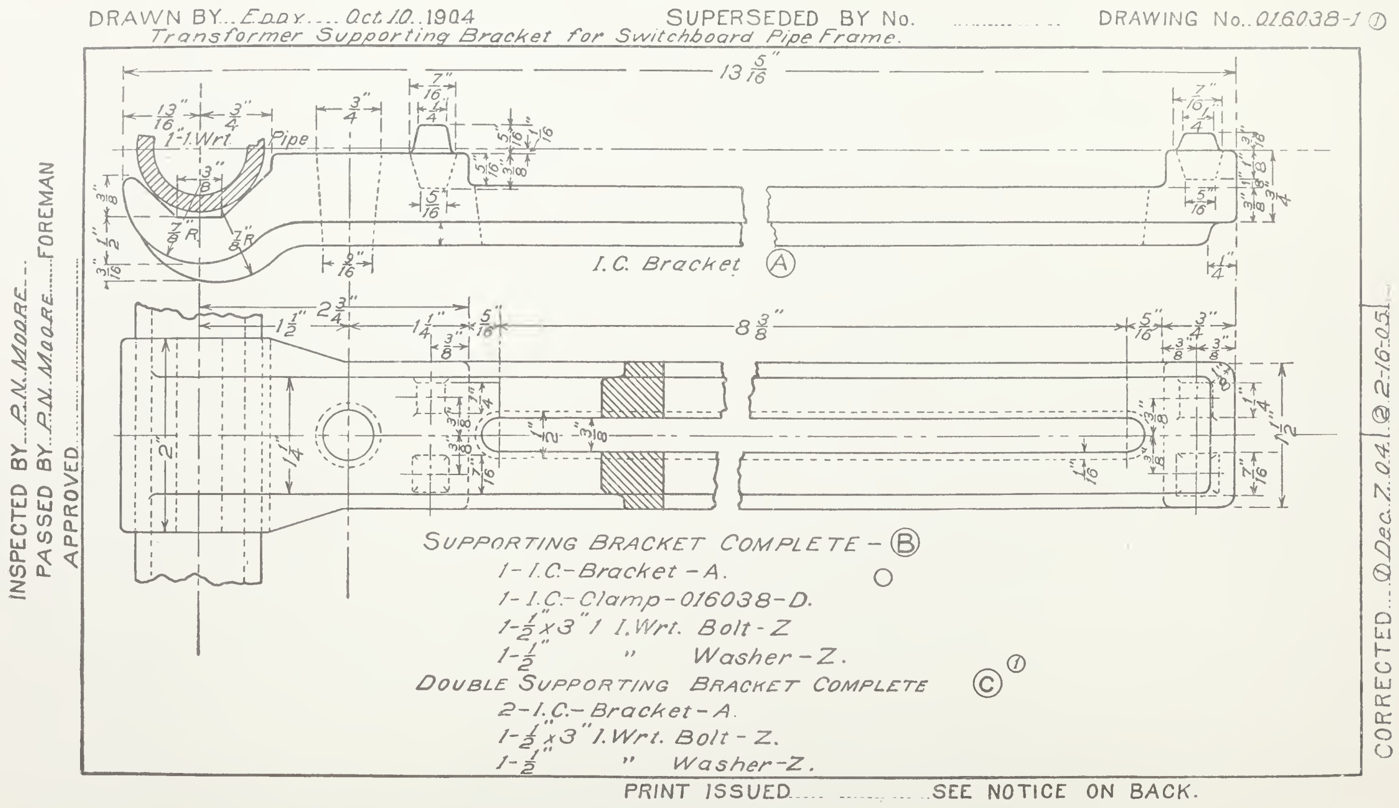
Detail Drawing. After the sketches are made, the next step is the making of the pencil drawing from the sketches, accurately to scale. The size of the plate on which the drawing is to be made is usually fixed by some standard. Where many drawings are made and kept in an office, it is desirable to keep the plates of uniform size, as far as possible. It is good practice to have two or three standard sizes of plates, one for small drawings, one for ordinary-sized work, and one for large drawings; then, whenever a drawing is made, make it on one of these standard plates.

Assuming, then, that we have our paper stretched on the drawing board and the plate laid out, the next step will be to arrange the drawings of the various pieces on the plate so that there will be room for all and so that they may be properly placed with relation to each other. It may happen that there will not be room on one plate for all the pieces, but that two or more plates will be required. When the parts must be thus arranged on different plates, an effort should be made to keep on the same plate those parts which belong together. For example, if we were drawing a lathe, the details of the parts of the head stock might form one plate, the apron another, and so on.

In locating the various pieces on a plate, they should be placed as nearly as possible in the same relative position to each other that they bear in the machine, except that they are separated. For example: if a nut belongs on the end of a screw, it is desirable to draw it on the same center line with the screw and at the end where it belongs. If a piece is vertical in the machine it should be vertical on the plate, and if horizontal in the machine, it should be horizontal on the plate.

The approximate location of the pieces on the plate may be easily decided by taking a small sheet of paper of about the same proportion as the plate, but perhaps one-fourth or one-half size, and sketching on it roughly the outline of the various pieces. The arranging of the plate should not be allowed to take much time, but should be done as rapidly as possible. After the location of each view of each piece is determined, the pencil drawing should begin (to scale) with one of the principal pieces. In almost all cases a center line is first drawn. It is better to carry along all the views of a piece at once, instead of completing one view at a time. The piece started should have all its views finished and completely dimensioned before another piece is begun; exceptions to this are sometimes necessary for special reasons. The lines should be drawn accurately, but no attempt need be made to obtain finish; that is, in order to save time, the lines may be run past the point where they should properly stop, etc. Nothing should be omitted, however.

Each plate of details should have a title, stating of what machine it is a drawing, and, if there are several plates, it is well to state also of what part of the machine the plate in question is a drawing. It is also a good idea to print its name beside each of the principal pieces.


DRAWN BY...Eddy.... Oct.10..1904
SUPERSEDED BY No.
DRAWING No..016038-1 ①
Transformer Supporting Bracket for Switchboard Pipe Frame.
13 5/16"
1"-1.Wrt. Pipe
7/8"R
I.C. Bracket Ⓐ
8 3/8"
2 3/4"
1 1/2"
1 1/4"
2"
SUPPORTING BRACKET COMPLETE - Ⓑ
1-I.C.-Bracket-A.
1-I.C.-Clamp-016038-D.
1-1/2"x3" I.Wrt. Bolt-Z
1-1/2" " Washer-Z.
DOUBLE SUPPORTING BRACKET COMPLETE Ⓒ ①
2-I.C.-Bracket-A.
1-1/2"x3" I.Wrt. Bolt-Z.
1-1/2" " Washer-Z.
PRINT ISSUED........ SEE NOTICE ON BACK.
INSPECTED BY...P.N. Moore...
PASSED BY...P.N. Moore......FOREMAN
APPROVED
CORRECTED...① Dec.7.04 ② 2-16-05

TYPICAL SHOP DRAWING MADE IN THE DRAFTING ROOM OF THE STANLEY ELECTRIC MFG. CO.

Tracing. Having finished the pencil drawing, the next step is the inking. In some offices the pencil drawing is made on a thin. tough paper, called board paper, and the inking is done over the pencil drawing, in the manner with which the student is already familiar. It is more common to do the inking on thin, transparent cloth, called tracing cloth, which is prepared for the purpose. This tracing cloth is made of various kinds, the kind in ordinary use being what is known as "dull back," that is, one side is finished and the other side is left dull. Either side may be used to draw upon, but most draftsmen prefer the dull side. If a drawing is to be traced it is a good plan to use a 3H or 4H pencil, so that the lines may be easily seen through the cloth.

The tracing cloth is stretched smoothly over the pencil drawing and a little powdered chalk rubbed over it with a dry cloth, to remove the slight amount of grease or oil from the surface and make it take the ink better. The dust must be carefully brushed or wiped off with a soft cloth, after the rubbing, or it will interfere with the inking.

The drawing is then made in ink on the tracing cloth, after the same general rules as for inking the paper, but care must be taken to draw the ink lines exactly over the pencil lines which are on the paper underneath, and which should be just heavy enough to be easily seen through the tracing cloth. The ink lines should be firm and fully as heavy as for ordinary work. In tracing, it is better to complete one view at a time, because if parts of several views are traced and the drawing left for a day or two, the cloth is liable to stretch and warp so that it will be difficult to complete the views and make the new lines fit those already drawn and at the same time conform to the pencil lines underneath. For this reason it is well, when possible, to complete a view before leaving the drawing for any length of time, although of course on views in which there is a good deal of work this cannot always be done. In this case the draftsman must manipulate his tracing cloth and instruments to make the lines fit as best he can. A skillful draftsman will have no trouble from this source, but the beginner may at first find difficulty.

Inking on tracing cloth will be found by the beginner to be quite different from inking on the paper to which he has been accustomed, and he will doubtless make many blots and think at first that it is hard to make a tracing. After a little practice, however, he will find that the tracing cloth is very satisfactory and that a good drawing can be made on it quite as easily as on paper.

The necessity for making erasures should be avoided, as far as possible, but when an erasure must be made a good ink rubber or typewriter eraser may be used. If the erased line is to have ink placed on it, such as a line crossing, it is better to use a soft rubber eraser. All moisture should be kept from the cloth.

Blue Printing. The tracing, of course, cannot be sent into the shop for the workmen to use, as it would soon become soiled and in time destroyed, so that it is necessary to have some cheap and rapid means of making copies from it. These copies are made by the process of blue printing in which the tracing is used in a manner similar to the use made of a negative in photography.

Almost all drafting rooms have a frame for the purpose of making blue prints. These frames are made in many styles, some simple, some elaborate. A simple and efficient form is a flat surface usually of wood, covered with padding of soft material, such as felting. To this is hinged the cover, which consists of a frame similar to a picture frame, in which is set a piece of clear glass. The whole is either mounted on a track or on some sort of a swinging arm, so that it may readily be run in and out of a window.

The print is made on paper prepared for the purpose by having one of its surfaces coated with chemicals which are sensitive to sunlight. This coated paper, or blue-print paper, as it is called, is laid on the padded surface of the frame with its coated side uppermost; the tracing laid over it right side up, and the glass pressed down firmly and fastened in place. Springs are frequently used to keep the paper, tracing, etc., against the glass. With some frames it is more convenient to turn them over and remove the backs. In such cases the tracing is laid against the glass, face down; the coated paper is then placed on it with the coated side against the tracing cloth.

The sun is allowed to shine upon the drawing for a few minutes, then the blue-print paper is taken out and thoroughly washed in clean water for several minutes and hung up to dry.

If the paper has been recently prepared and the exposure properly timed, the coated surface of the paper will now be of a clear, deep blue color, except where it was covered by the ink lines, where it will be perfectly white.

The action has been this: Before the paper was exposed to the light the coating was of a pale yellow color, and if it had then been put in water the coating would have all washed off, leaving the paper white. In other words, before being exposed to the sunlight the coating was soluble. The light penetrated the transparent tracing cloth and acted upon the chemicals of the coating, changing their nature so that they became insoluble; that is, when put in water, the coating, instead of being washed off, merely turned blue. The light could not penetrate the ink with which the lines, figures, etc., were drawn, consequently the coating under these was not acted upon and it washed off when put in water, leaving a white copy of the ink drawing on a blue background. If running water cannot be used, the paper must be washed in a sufficient number of changes until the water is clear. It is a good plan to arrange a tank having an overflow, so that the water may remain at a depth of about 6 or 8 inches.

The length of time to which a print should be exposed to the light depends upon the quality and freshness of the paper, the chemicals used and the brightness of the light. Some paper is prepared so that an exposure of one minute, or even less, in bright sunlight, will give a good print and the time ranges from this to twenty minutes or more, according to the proportions of the various chemicals in the coating. If the full strength of the sunlight does not strike the paper, as, for instance, if clouds partly cover the sun, the time of exposure must be lengthened.

Assembly Drawing. We have followed through the process of making a detail drawing from the sketches to the blue print ready for the workmen. Such a detail drawing or set of drawings shows the form and size of each piece, but does not show how the pieces go together and gives no idea of the machine as a whole. Consequently, a general drawing or assembly drawing must be made, which will show these things. Usually two or more views are necessary, the number depending upon the complexity of the machine. Very often a cross-section through some part of the machine, chosen so as to give the best general idea with the least amount of work, will make the drawing clearer.

The number of dimensions required on an assembly drawing depends largely upon the kind of machine. It is usually best to give the important over-all dimensions and the distance between the principal center lines. Care must be taken that the over-all dimensions agree with the sum of the dimensions of the various details. For example, suppose three pieces are bolted together, the thickness of the pieces according to the detail drawing, being one inch, two inches, and five and one-half inches respectively; the sum of these three dimensions is eight and one-half inches and the dimensions from outside on the assembly drawing, if given at all, must agree with this. It is a good plan to add these over-all dimensions, as it serves as a check and relieves the mechanic of the necessity of adding fractions.

FORMULA FOR BLUE-PRINT SOLUTION.

Dissolve thoroughly and filter.

A.	Red Prussiate of potash	2½ ounces,
	Water	1 pint.
B.	Ammonio-Citrate of iron	4 ounces,
	Water	1 pint.

Use equal parts of A and B.

FORMULA FOR BLACK PRINTS

Negatives. White lines on blue ground; prepare the paper with

Ammonio-Citrate of iron	40 grains,
Water	1 ounce.

After printing wash in water.

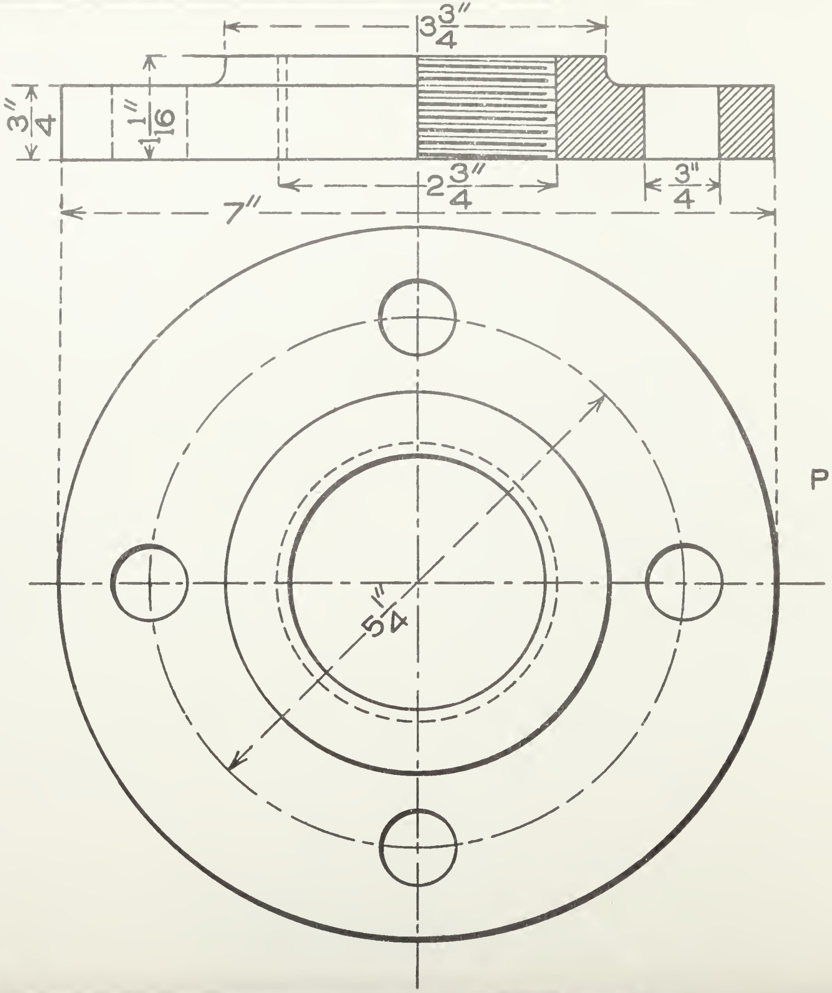
Positives. Black lines on white ground; prepare the paper with:

Iron perchloride	616 grains,
Oxalic Acid	308 grains,
Water	14 ounces.

Develop in	Gallic Acid	1 ounce,
	Citric Acid	1 ounce,
	Alum	8 ounces.

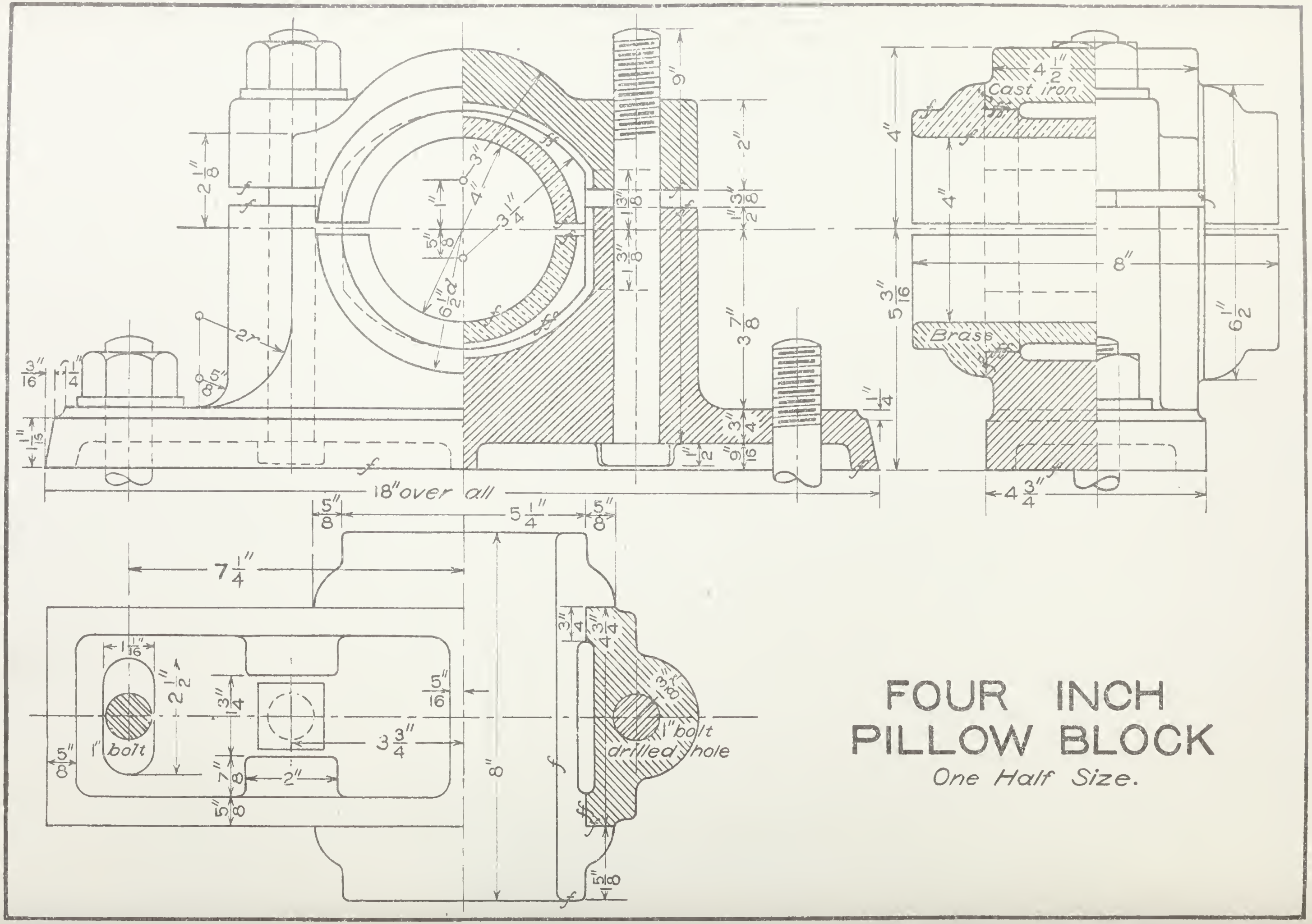
Use 1¼ ounces of developer to one gallon of water. Paper is fully exposed when it has changed from yellow to white.

PLATE XVII



PIPE COUPLING

APRIL 14, 1901 HERBERT CHANDLER CHICAGO ILL.



FOUR INCH PILLOW BLOCK

One Half Size.

APRIL 27, 1901   HERBERT CHANDLER CHICAGO, ILL.

PLATES.

PLATE XVI.

Pipe Coupling. The drawing represents one of a pair of cast-iron couplings used for connecting two lengths of pipe. It is threaded for a right-handed thread, but appears left-handed because the back part of the thread is visible in the section.

The vertical center line of the coupling is in the center of the plate; the center of the plan is $3\frac{7}{8}$ inches from the lower border and the bottom of the elevation $1\frac{7}{8}$ inches from the top border line. The elevation is made in half section, that is, the right-hand half. It is made by imagining that the front right quarter of the coupling is cut away by vertical planes. If the cutting plane passes through a hole or opening the cross-hatching is omitted. The dotted lines at the left of the hole show that it is threaded. The distance between these lines is $\frac{3}{32}$ inch, the depth of the thread. The threaded or tapped portion is shown in plan by the two circles, the dotted circle representing the outside and the full circle the inside or root of the thread.

The boss or hub adds to the strength and allows more threads. The shade lines should be placed on the drawing following the custom for shop drawings. The title "Pipe Coupling" should be placed as shown and made with letters $\frac{1}{4}$ inch high. The letters may be either vertical or inclined Gothic capitals.

PLATE XVII.

Pillow Block. The drawings for this plate are three views—side, plan and end—of a 4 inch-pillow block. Each view is half in section. The drawing should be one-half size. Make the vertical center line of the side and plan views $4\frac{3}{4}$ inches from the left-hand border. The horizontal center line of the side view should be $2\frac{3}{8}$ inches from the top border line and the horizontal center line of the plan $2\frac{1}{4}$ inches from the bottom border line. Locate the vertical center line of the end view $2\frac{1}{4}$ inches from the right-hand border line; the horizontal center line will of course be a continuation of that of the side view. Note that the left-hand half of the plan represents what is seen by looking up from below. The cutting plane for the right-hand half passes horizontally through the center of the shaft.

The student should put on shade lines and finish marks; also the notes and dimensions. Make the letters of the title vertical Gothic capitals $\frac{3}{8}$ inch high.

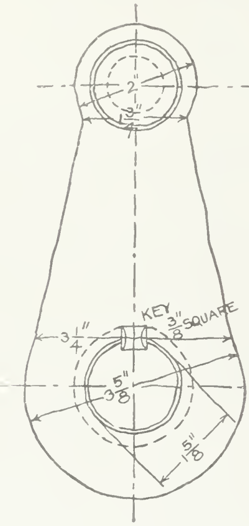


Fig. 48.

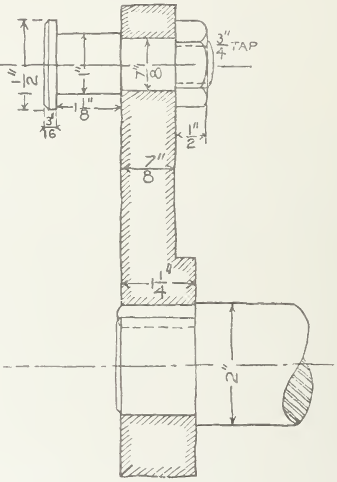


Fig. 49.

PLATE XVIII.

Overhung Crank. The two drawings on this plate are to be made full size from the free-hand sketches Figs. 48 and 49. These are such as would be given the draftsman or such he would make for his own work. Place the two views so that they will look well on the plate. Put on shade lines and dimensions.

PLATE XIX.

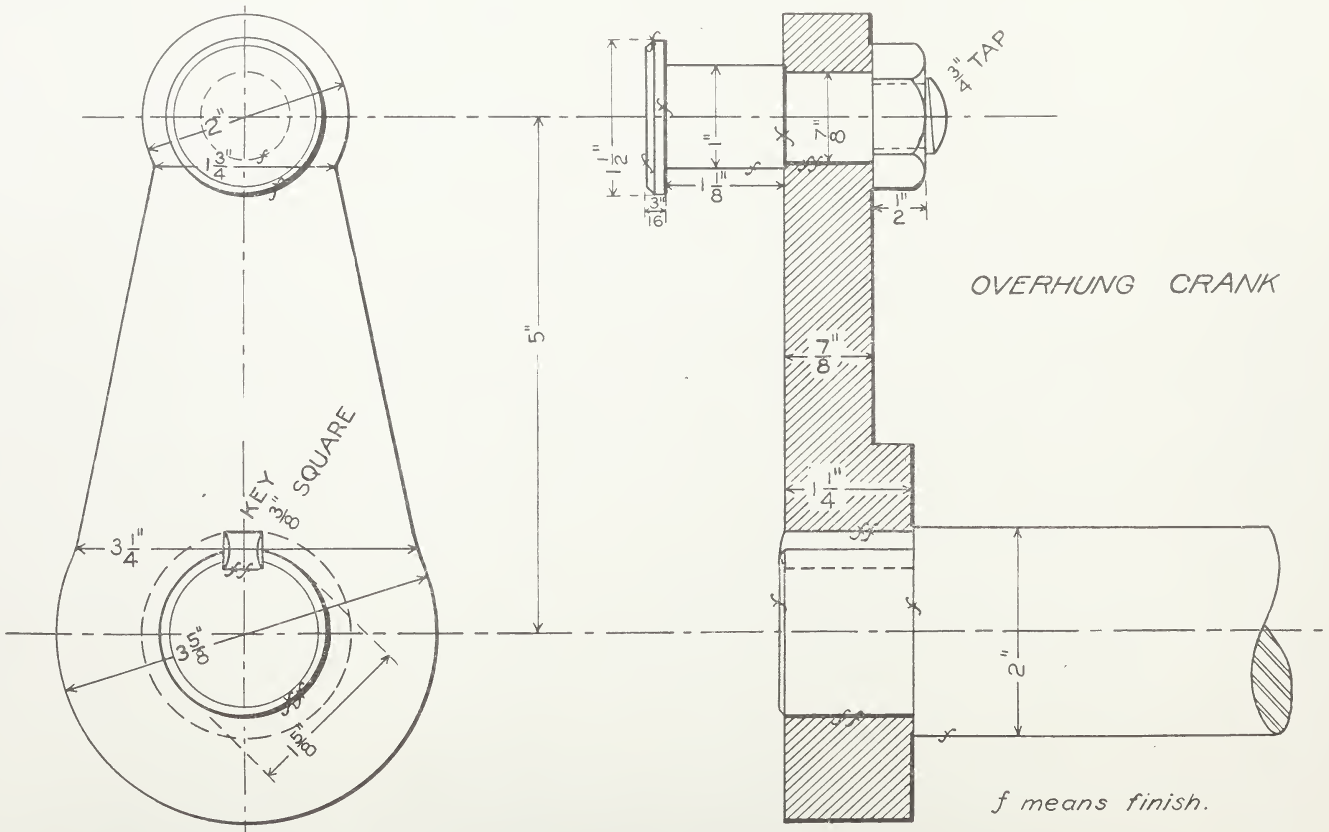
Cylinder Head. This plate consists of a plan view and a cross-section of the cylinder head of a small engine. The center line for the two views is drawn half way between the upper and lower border lines. Always allow sufficient space for dimensions.

The shade lines should be placed on the drawing, following the methods described for machine drawing. Locate the title and explanatory notes to make the plate appear well balanced.

PLATE XIX A.

Blue Print. Make a tracing on tracing cloth and (optional) a blue print of one of these plates.

PLATE XVIII.



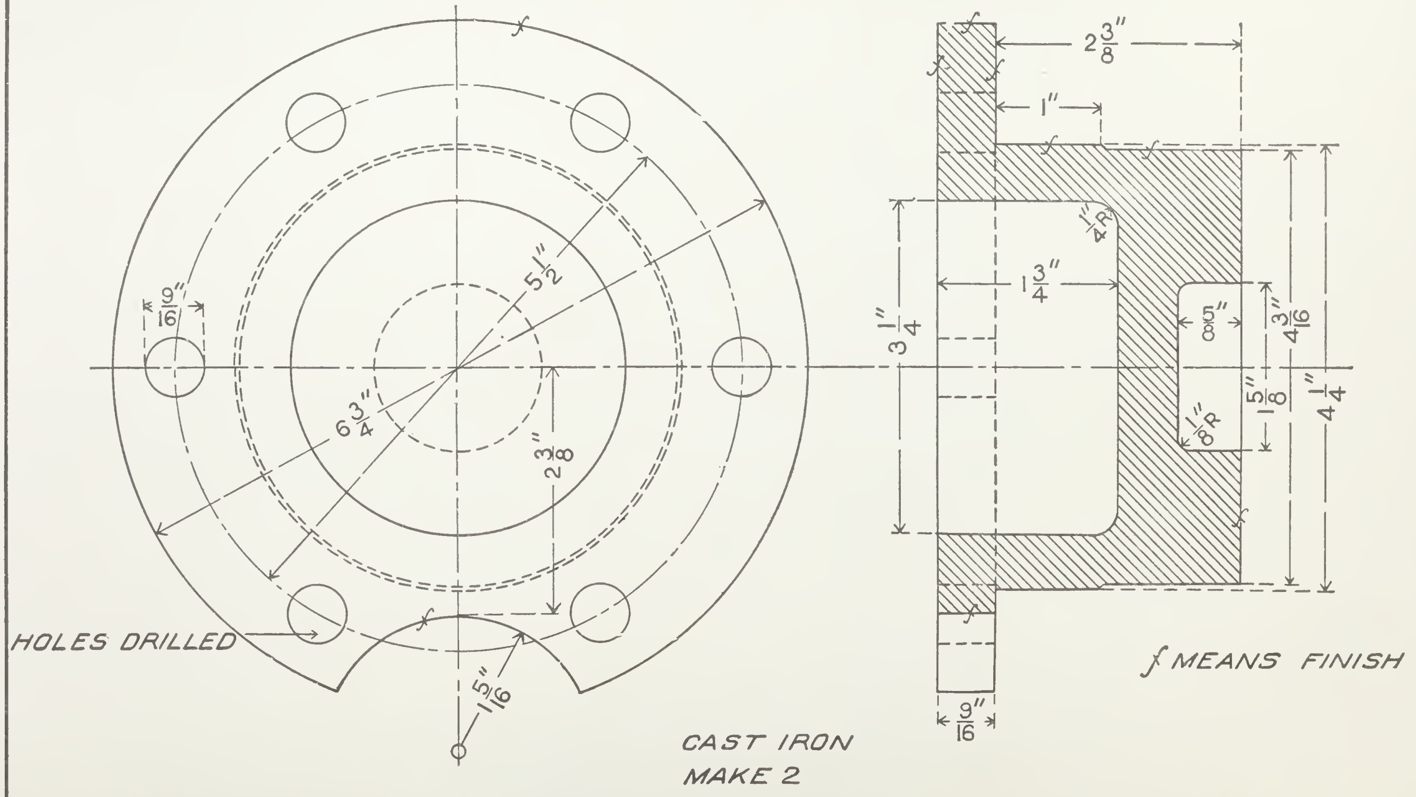
OVERHUNG CRANK

f means finish.

MAY 12, 1901. HERBERT CHANDLER, CHICAGO, ILL.

PLATE XIV

CYLINDER HEAD



MAY 20, 1901. HERBERT CHANDLER CHICAGO, ILL.

MECHANICAL DRAWING.

PART V.

MACHINE DRAWING.

A machine consists of a number of parts, or groups of parts, so arranged and connected that, when power is applied at one part, definite motions and forces are transmitted to the other parts, which in turn do certain definite work. The parts of a machine may be divided in a general way into two classes: first, those parts which have or cause motion; and second, those which act as supports for the moving parts, such as frame, bearings, etc.

One of the groups of parts of a machine, arranged to accomplish some one particular result, is called a mechanism. Most machines are composed of several mechanisms, one doing one thing and another doing something else, and the whole working together to do the work for which the machine is designed. Take, for example, a metal planer; the gears, levers, etc., which cause the platen to move back and forth form one mechanism; the parts which feed the tool across form another mechanism, and so on, all the mechanisms taken together forming the machine.

Among the most common parts of machines are springs, screws, cams, pulleys and belts, and gears, and consequently it is necessary that a draftsman should understand how to draw these parts.

THE HELIX.

Since most coil springs and all screw threads depend upon a curve known as a helix, it will be necessary to know what a helix is, and how it can be drawn, before taking up the construction of the springs and screws.

Suppose we take a cylindrical piece of wood, such as is shown in Fig. 1, and a rectangular piece of paper, A B F E, with the side A B equal to the circumference of the cylinder, and the side A E equal to the length of cylinder. If we lay off along A E any convenient distance, A D, and draw the line D C parallel to A B, we have the rectangle A B C D. Now draw the diagonal, A C, of this rectangle and wrap the paper around the cylinder, keeping the side A E on an element of the cylinder; the paper will just cover the cylinder, the edge B F meeting the edge A E. The point C coincides with point D, and is on the same element of the cylinder as A; therefore the line A C has made one complete turn around the cylinder, advancing the distance A D in this turn. The curve which the line A C now takes is called a helix, and the distance A D is called the pitch of the helix.

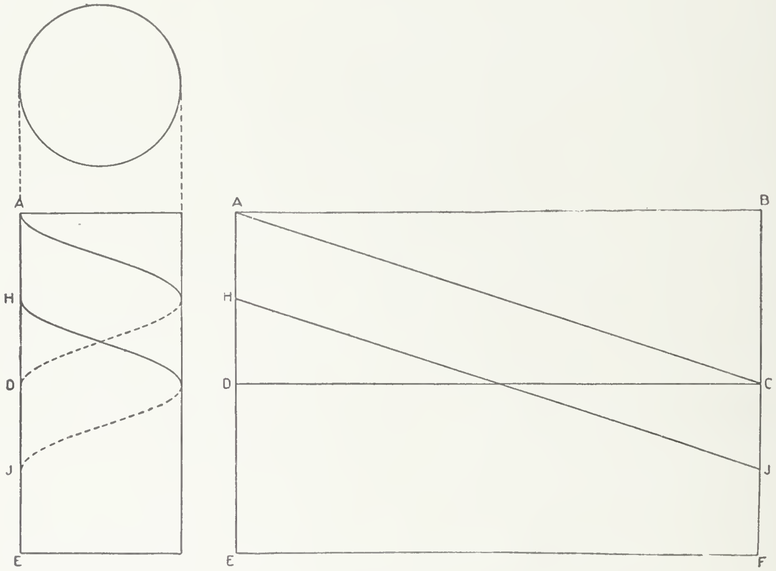


Fig. 1.

If on the piece of paper we also choose a point, H, halfway between A and D, and draw from this point a line, H J, parallel to line A C, this line H J will form another helix parallel to the helix found by the line A C, when the paper is wrapped around the cylinder. The pitch of both helices is the same.

The helix is often incorrectly called a spiral, but there is a marked difference between the two; the spiral is a curve which lies in one plane, and winds around a point, constantly receding from the point, according to some law. The mainspring and hairspring of a watch are in the form of spirals.

To draw the projections of a helix we must know the diameter of the cylinder about which the helix is wound, and the pitch of the helix. Fig. 2 shows the construction. Draw two projections of the cylinder A B D C ; along any element, preferably one of the contour elements, A B or C D, lay off the pitch A E. Divide the circumference of the circle, which is the end view of the cylinder, into any number of equal parts, and number the points of division 1, 2, 3, etc. Divide the pitch A E into the same number of equal parts, and number these points of division in the same way that the points on the circle are numbered, calling the point A No. 1. From point 2 on the circle draw a line parallel to A B, and from point 2 on A B draw a perpendicular to A B. The point, L, where the parallel line meets the perpendicular line, is one point on the projection of the helix. The points M, N, etc., are found in the same manner. A smooth curve starting from A, going through all the points and ending at E, will be the projection of one turn of the helix. The half from A to R is on the front, and is therefore a full line, while the half from R to E is on the back and is a dotted line. It should be observed that the point R is on the perpendicular from 7, which is just halfway between A and E, that is, the distance C R is just one-half the pitch. The curve from E to B is the projection of the next turn of the helix and is exactly like the first one.

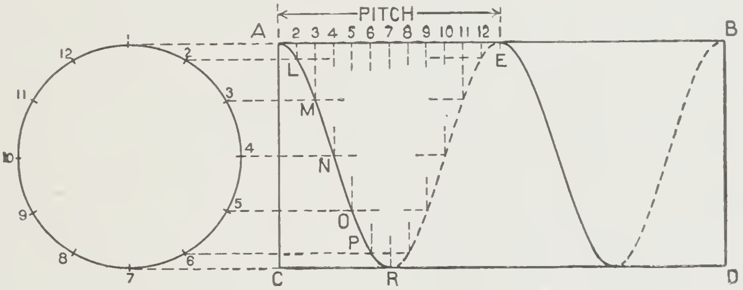


Fig. 2.

The helix shown in Fig. 2 is called a right-handed helix. If the curve starts at C and is drawn as in Fig. 3, we have a left-handed helix. Notice that the visible part (from C to R) slants in the same direction as the invisible part of the right-handed helix, which is shown dotted in Fig. 2.

Since the helix is a line drawn on the surface of a cylinder, the other projection of the helix must be the circumference of the circle, which is the end view of the cylinder. Fig. 4 shows a right-handed double helix, and Fig. 5 is a right-handed triple helix.

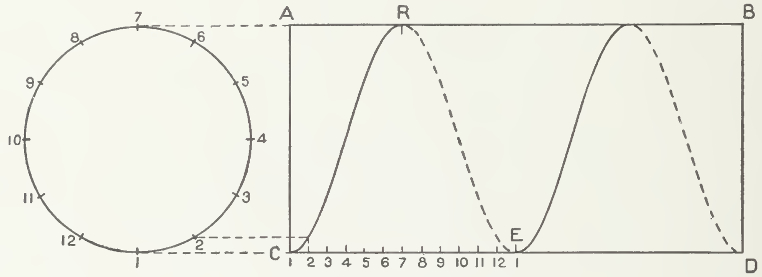


Fig. 3.

The construction of these curves should be studied carefully in order that the springs and screw threads may be better understood.

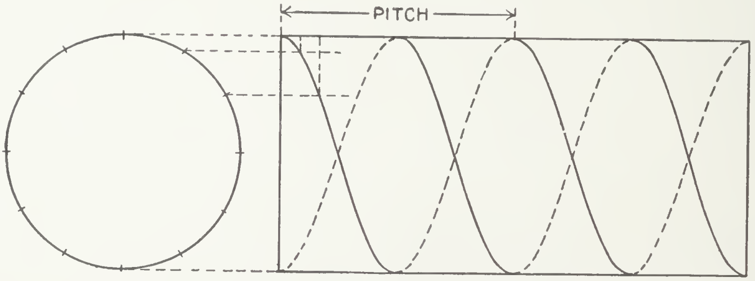


Fig. 4.

Helical Springs. If, instead of winding a line around a cylinder in the form of a helix, as shown in the preceding figures, we wind a piece of spring wire, we will get a helical spring.

Fig. 6 is the drawing of a helical spring of round wire, the side view only being drawn, as this is all that is necessary to give all the dimensions. To draw the spring we must know the pitch, the diameter of the wire, and either one of the dimensions, A, B or C. If A is given, we subtract from A the diameter of the wire to find B; and if C is given, we add to C the diameter of the wire to get B; then knowing B and the pitch we can draw the helix (which is shown in dotted lines) exactly as the helix was drawn in Fig. 2. This is the helix formed by a line in the center of the wire. Now draw a series of circles with centers on this helix and of a diameter equal to the diameter of the wire. Smooth curves drawn tangent to these circles, as shown in the figure, will give the projection of the spring.

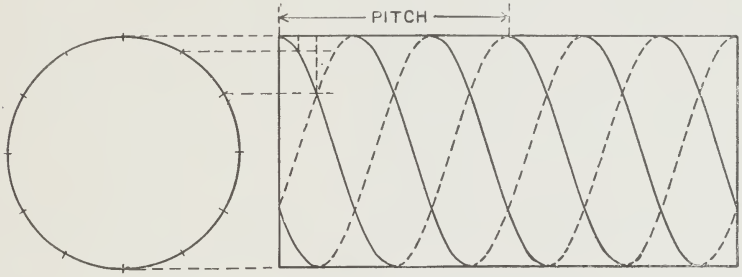


Fig. 5.

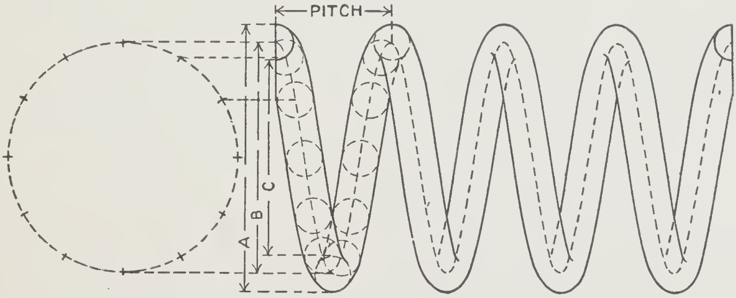


Fig. 6.

Fig. 7 shows a helical spring of square wire. The drawing of this is simply the drawing of four helices, starting from each of the corners of the square, A C M L; this square being the cross-section of the wire of which the spring is made. All four of the helices have the same pitch, equal to A B, for since the square

B D P N is the same as A C M L, the distance C D is the same as A B; and since the points L, M, N and P are vertically under A, C, B and D, respectively, the distance L N is equal to A B, and M P is equal to C D. The helix A F B has a diameter equal to that of the circle I E and is drawn by dividing the circle I E and the pitch A B, as in Fig. 2; and the helix C G D, having the same diameter as A F B, is drawn by dividing circle I E and pitch C D. The helix L H N has a diameter equal to that of the circle K R, which is I E minus twice the diameter of the wire, and is drawn by dividing up the circle K R and the pitch L N; and the helix M J P, having the same diameter as L H N, is drawn by dividing circle K R and pitch M P. Since the two circles are drawn about the same center, the divisions on circle K R can be found by drawing radial lines from the points of division on circle

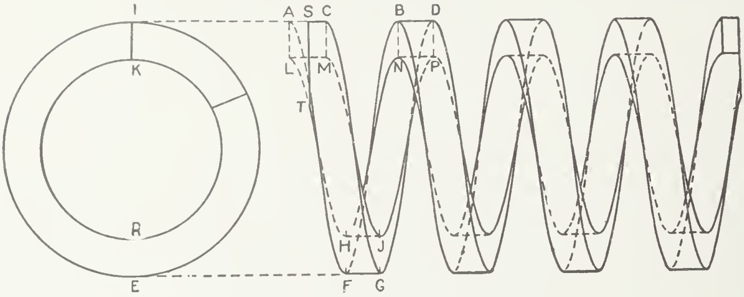


Fig. 7.

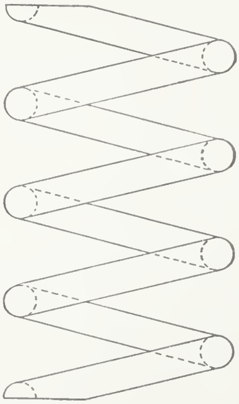


Fig. 8.

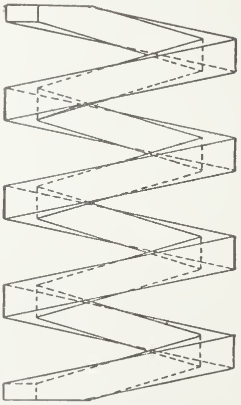


Fig. 9.

I E. The vertical lines drawn from the divisions of the pitch A B can be used for the divisions of L N, and those drawn from divisions of C D can be used for M P.

After the four parallel helices are drawn it is necessary to study the drawing carefully to decide what lines will be visible (full lines) and what invisible (dotted lines). Dotted lines should be used from H to J, N to P, etc., and full lines from F to G, B to D, etc. The line S T is the end of the spring, and consequently any part of a helix which goes outside of that line should not be left on the finished drawing. It is better, however, to draw in the whole of the square A C M L and to draw the helices starting from A and L in order to draw those parts of the same helices which lie to the right of S T. The parts to the left of S T are shown in the figure by light dotted lines to indicate that they are construction lines, and not a part of the projection of the spring itself.

Conventional Representations of Springs. To draw springs by the method just explained involves considerable work and would consume a great deal of time if many were to be drawn, therefore in working drawings the draftsman frequently uses a conventional method. This conventional drawing is similar to the true projection, except that straight lines are used in place of curved lines. Fig. 8 shows the conventional drawing of a spring of round wire, and Fig. 9 of a spring of square wire. Springs are often shown in half-section, as in Fig. 10, this method involving less work than the method of Figs. 8 and 9.

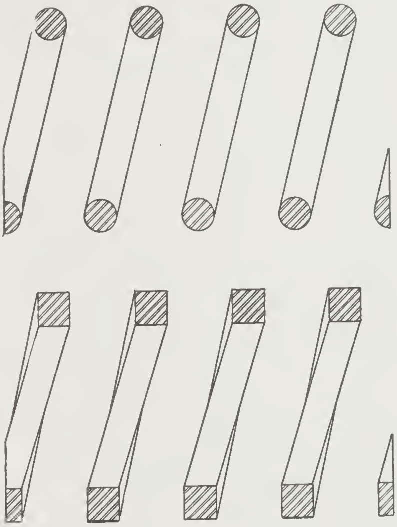


Fig. 10.

SCREW THREADS.

If we cut a groove around a cylinder in the form of a helix we shall have what is called a screw thread, the thread being

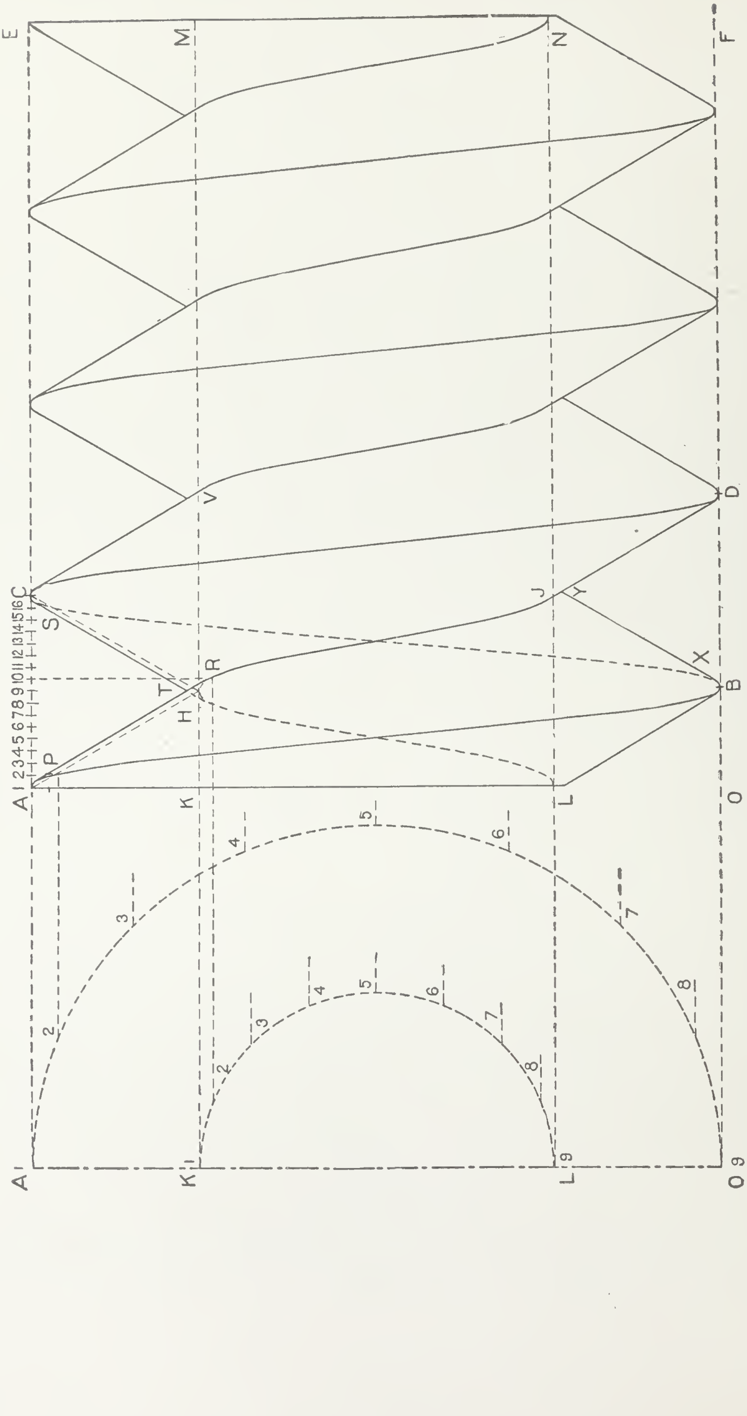
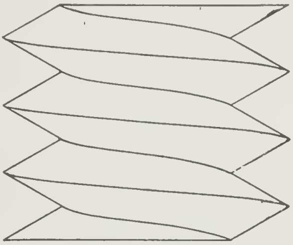
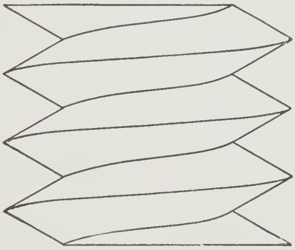


Fig. 12. Construction of V Thread.

formed by the material which is left between the successive turns of the helical groove. A cylinder having such a helical groove cut around it is called a screw, and a piece having a cylindrical hole in it with a helical groove cut around the hole is called a nut. The most common uses of the screw are to fasten pieces together or hold them at a given distance apart, and to cause one piece to move with relation to another piece.



LEFT-HAND THREAD.



RIGHT-HAND THREAD.

Fig. 11.

V Thread. The form of screw thread with which we are most familiar is what is known as the V thread, shown in its simplest form in Fig. 11. Fig. 12 shows the method of drawing the true projections of this thread. The dimensions which must be known in order to make the drawing are the outside diameter, A O, the pitch, A C, and the depth of the thread, A K. First draw the two projections of a cylinder of a diameter equal to the outside diameter of the screw. Half of the end view is sufficient. On the line A E of this cylinder lay off A C equal to the pitch, and starting at A draw the helix A B C D, as described for Fig. 2. Inside of the cylinder A O, draw a smaller cylinder, K L, the diameter of which is equal to the diameter A O minus twice the depth of the thread. Now on this smaller cylinder, starting at point H, perpendicularly under a point on the line A C which is halfway from A to C, draw the helix H J V with the same pitch as was used for the helix A B C. Draw the lines P R, X Y, S T, etc., tangent to the two helices, and the projection of the thread is completed. It is necessary to draw the invisible parts of the two helices in order to draw the lines S T, X Y, etc., but they need not be left on the finished drawing. In Fig. 12 they are shown dotted for one turn of the screw in order to show the construction.

Fig. 13 shows the method of drawing a double V thread.

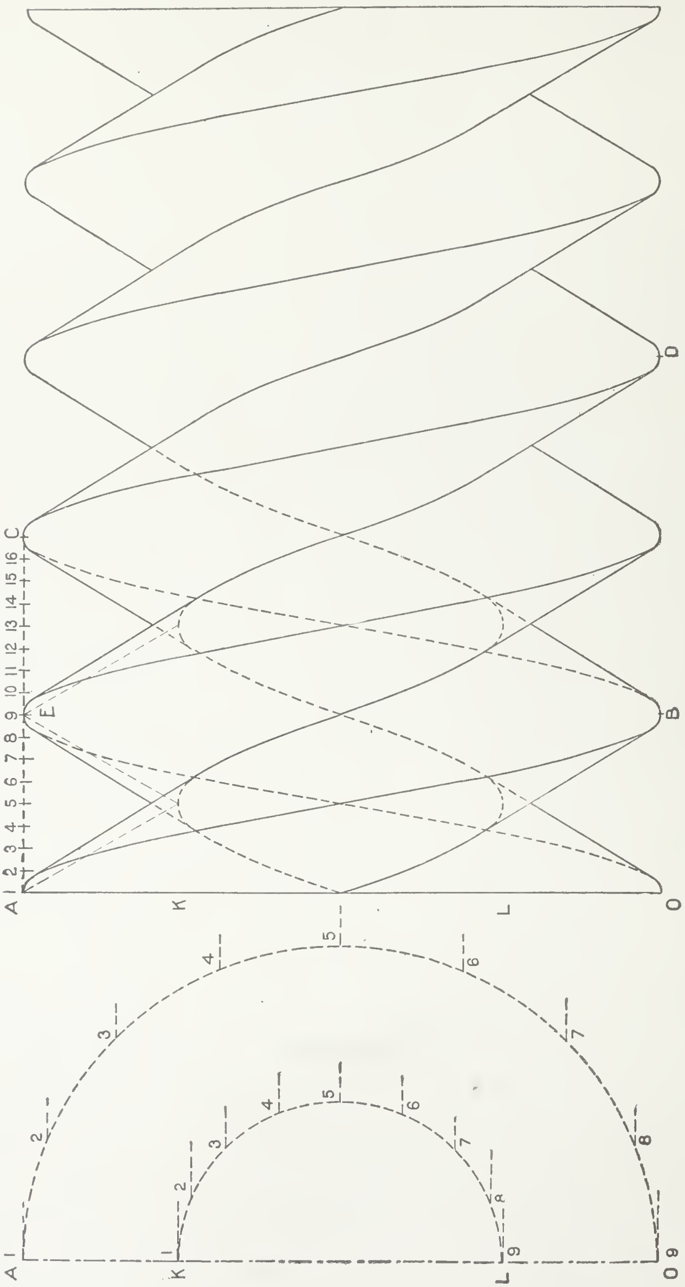
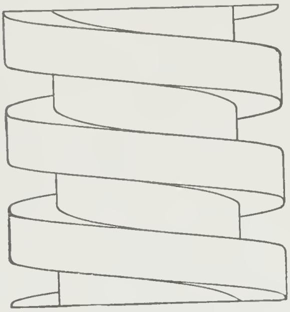
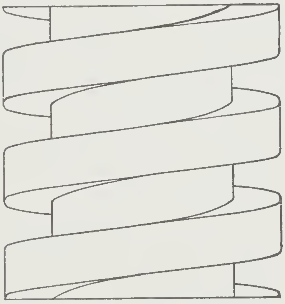


Fig. 13. Construction of Double V Thread.

The process is exactly the same as for drawing a single thread. Start at point A and draw the single thread A B C D exactly as in Fig. 12; then start at point E, halfway between A and C, and draw another single thread of the same pitch as the first one. Some thought may be necessary to decide when the lines of one thread become hidden behind the other thread.



LEFT-HAND THREAD.



RIGHT-HAND THREAD.

Fig. 14.

Square Thread. Another very common form of screw thread is that shown in Fig. 14, and is known as the square thread. The method of drawing this thread is similar to that for the V thread with the exception of a few minor points. The construction is shown in Fig. 15. The dimensions which must be known are the outside diameter, A O, the pitch, A C, the depth, A H, and either the width of the thread, A R, or the width of the groove, R C. In the figure, the width of the thread, A R, is taken equal to one-half of the pitch, that is, A R and R C are equal. Beginning at A, draw the helix A B C, and beginning at R draw the helix R M N, R N of course being equal to A C. Since the part between A and R is metal, forming the thread, there will be a line from A to R and from B to M, etc. Now, starting at point H vertically under A and at a distance from A equal to the depth of the thread, draw the helix H J V, and from S vertically under R draw helix S T W. Draw the lines S V, T N, etc. Here, as in the case of the V thread, the invisible lines must be drawn when making the drawing; but need not be inked.

Fig. 16 shows the construction of a double square thread. An explanation is not necessary, since the difference between this

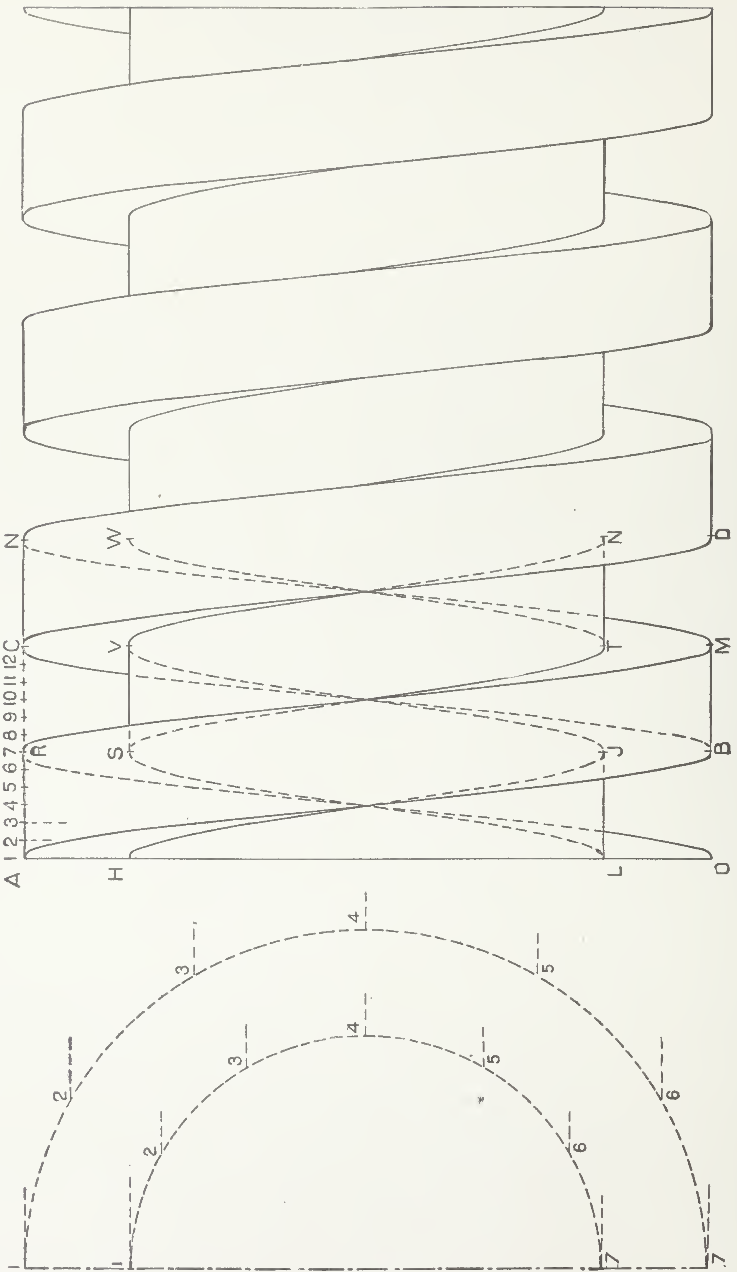


Fig. 15. Construction of Square Thread.

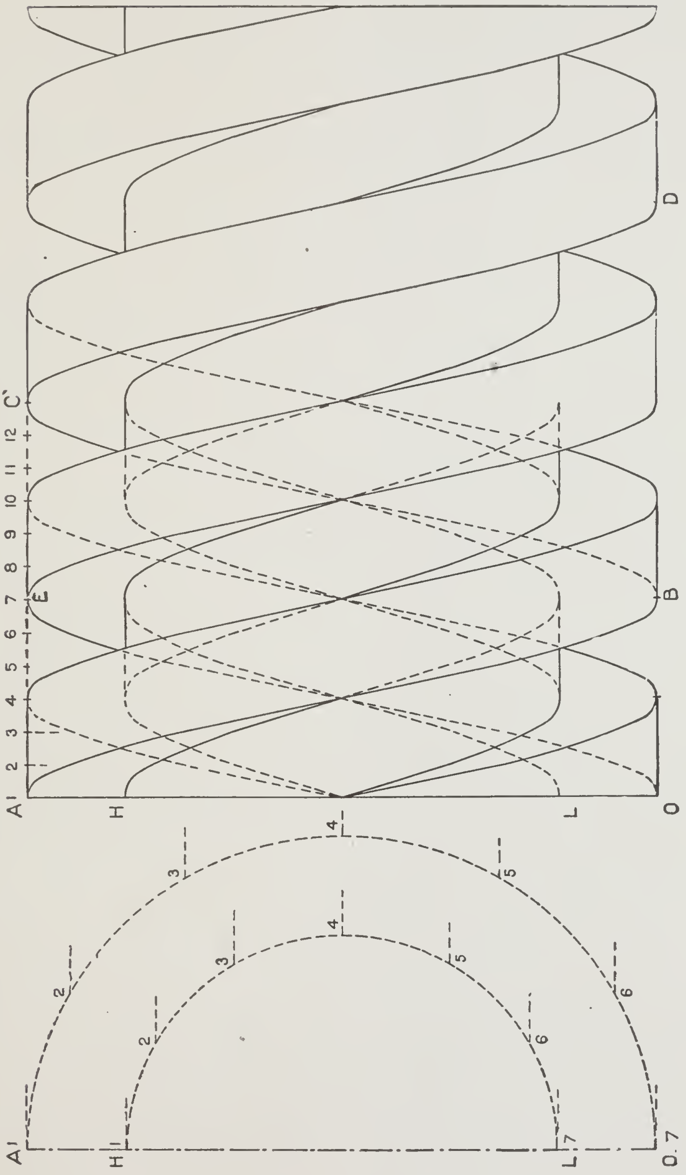


Fig. 16. Construction of Double Square Thread.

and the single square thread is practically the same as between the single and double V thread.

The V and square threads are the two fundamental forms of threads in use and all other forms are modifications of one or the other of these two. Figs. 17 to 20 show some of the more common modifications.

Figs. 17 and 18 show the two forms of the V thread which are commonly used in practice.

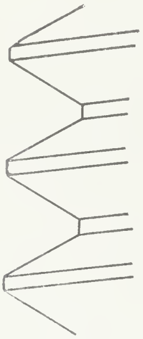


Fig. 17.

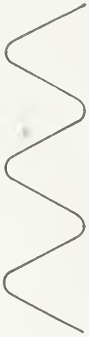


Fig. 18.

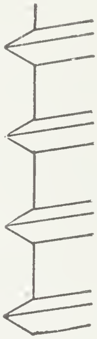


Fig. 19.

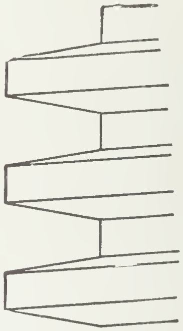


Fig. 20.

In Fig. 17 we have what is known as the Sellers or United States standard thread, an enlarged drawing of which is shown in Fig. 21. Referring to this figure, we see that the angle between the two sides of the thread is 60 degrees, so that if the thread came to a point at the top and bottom, as indicated by the dotted lines, the depth of the thread, D, would be about $\frac{87}{100}$ of the pitch, P. The sharp corners, however, are a disadvantage, since on the outside they are likely to be bruised, and give trouble in putting on the nut, and at the bottom of the groove they tend to weaken the bolt or screw. In order to avoid these sharp corners, the threads are flattened in the United States standard thread, as shown in Fig. 21, the amount of this flattening being such that the distance C is $\frac{1}{8}$ of the pitch, or, what amounts to the same thing, the distance A is $\frac{1}{8}$ of D. This gives a thread whose depth, E is $\frac{65}{100}$ of the pitch.

Fig. 18 illustrates what is known as the Whitworth standard thread, shown enlarged in Fig. 22. Here the angle between the sides of the thread is 55 degrees, so that if the threads came to sharp corners, as shown by the dotted lines, the depth D would be $\frac{96}{100}$ of the pitch. The top and bottom of the thread, instead of being flattened, are rounded off so that the distance A is $\frac{1}{6}$ of D, or the depth E is $\frac{64}{100}$ of the pitch.

Fig. 19 shows the V thread as used on lag screws and other wood screws. Here the groove is much larger than the thread, because the wood into which it is to screw is weaker than the iron of which the screw is made.

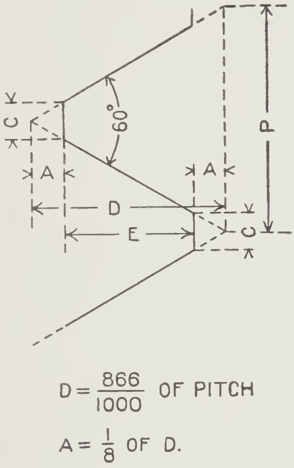


Fig. 21.

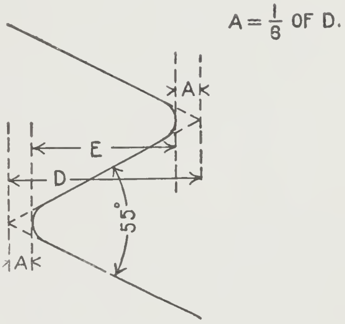


Fig. 22.

Fig. 20 shows a slightly modified form of square thread, the only difference between this and the square thread previously described being that the sides of the groove taper slightly.

Conventional Representations. The student should understand the drawing of the threads, as previously explained, and every draftsman should be able to draw the true projection of a thread if he should have occasion to do so. It is evident, however, that the process is somewhat long, and on a screw of small diameter and pitch it would be difficult to follow out the construction. To avoid this labor the construction shown in Figs. 23, 24 and 25 may be adopted, straight lines being substituted for the projections of the helix. Fig. 23 shows the conventional representation of the plain single right-hand V thread, the true projection of which was shown in Fig. 12. To make the conventional drawing (Fig. 23) draw the parallel lines A B and C D at a distance apart equal to the outside diameter of the screw, and draw the line A C perpendicular to these two lines. Along A lay off the distances A E, E F, etc., each equal to the pitch. Along C D lay off C H equal to $\frac{1}{2}$ the pitch, and from H lay off H I, I J, etc., equal to the pitch. Draw lines from A to H, E to I, etc. Now, if the depth of the thread A S is known, draw the lines S T and U V, and beginning at L, perpendicularly under a point halfway between A and E lay off L M, M N, etc., equal to the pitch. In like manner find the points O, P, R, etc., and draw the lines L O, M P, etc.; also A L, L E, H V, H O, etc. The dotted lines should be left out in the finished drawing, but are put in the figure to show the construction.

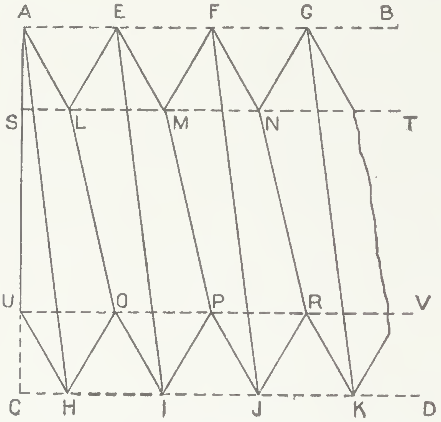


Fig. 23.

If, instead of knowing the depth A S, we know the angle between A L and L E, the depth can be found by drawing from A and E the two lines A L and E L in such a way that they would make the required angle with each other. To do this the lines A L and E L would each make an angle with the line A B equal to 90 degrees minus $\frac{1}{2}$ the angle between A L and L E.

Fig. 24 shows the corresponding construction for the United States standard thread. Draw the lines A B, C D and A C as in Fig. 23, and find the points E, F, G, H, I, J, K, etc., in the same way as in that case. Now draw the lines S T and U V so that A S and C U shall equal $\frac{65}{100}$ of the pitch A E. On the line A B lay off from A a distance, A 1, equal to $\frac{1}{16}$ of the pitch, and each side of E, F, G, H, etc., lay off E 2, E 3, etc., each equal to $\frac{1}{16}$ of the pitch. From the points thus found draw lines 1–4, 2–5, etc., making an angle of 60 degrees with A B. The rest of the drawing is completed as shown by drawing in full lines those parts of the lines A B, C D, S T and U V intercepted between 1–4 and 2–5, etc.

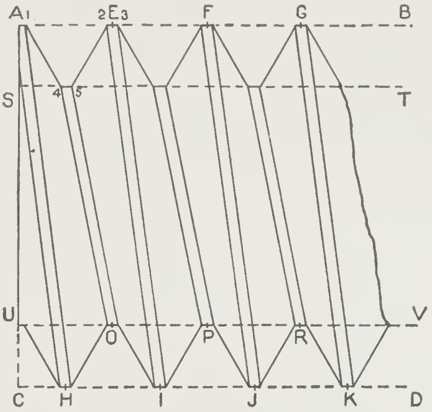


Fig. 24.

Fig. 25 shows the conventional representation of a square thread, and is drawn in exactly the same way as the true projection shown in Fig. 15, except that straight lines are used instead of curves.

Square threads are almost always represented as in Fig. 25, and V threads of coarse pitch and large diameter are shown as in Fig. 23 or 24, but when we have a screw of small diameter and fine pitch, such a method would become difficult, so that we most frequently find V threads represented as in Fig. 26. The lines A B and C D are drawn full, their distance apart being equal to the outside diameter of the screw. A F, F H, etc., are laid off to represent the pitch (though in small screws it is frequently made larger than the actual pitch). For a single thread, C E is equal to $\frac{1}{2}$ of A F, or, in other words, E is perpendicularly opposite a point halfway between A and F. A E is drawn, and F G, H J, etc., drawn parallel to A E. Halfway between the lines A E and F G, the shorter and heavier lines are drawn, the distance from AB and C D to the ends of the heavy lines being a little less than A F. To have these lines of uniform length it is well to draw light pencil lines, as a guide for stopping the heavy lines, these guide lines being erased when the drawing is finished.

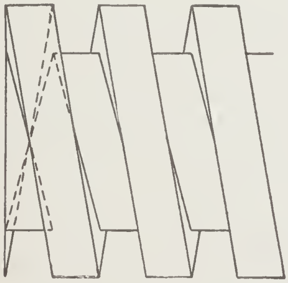


Fig. 25.

Fig. 27 shows the same method of representing a double thread. The only difference between this and Fig. 26 is that in Fig. 27 C E is equal to A F; that is, E is perpendicularly opposite F.

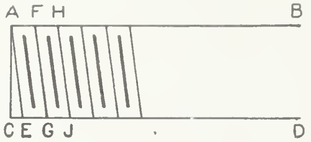


Fig. 26.

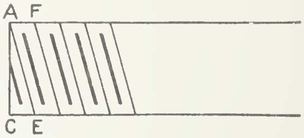


Fig. 27.

CAMS.

A cam is a piece of mechanism which, by means of its rotation about a fixed axis, imparts a back and forth motion to another piece in contact with it. There are several forms of cams which we shall consider, but before beginning the study of the cams and how to draw them it will be necessary for us to understand something of the different kinds of motion a piece may have, because a cam must be designed to give the piece the kind of motion desired.

Kinds of Motion. If we classify the kinds of motion which we meet with in machinery according to the rate of motion, or velocity, the three most common kinds are *uniform motion*, *harmonic motion* and *uniformly accelerated and uniformly retarded motion.* These will be more readily understood after we have considered one or two illustrations.

Uniform Motion. Suppose a railway train to travel ten miles in twenty minutes, the speed of the train being the same during each minute; it would travel $\frac{1}{2}$ mile in each minute, and would be said to have a *uniform* speed, or *uniform* velocity of $\frac{1}{2}$ mile per minute. Under such conditions the train would be moving at the full speed of $\frac{1}{2}$ mile per minute both at the beginning and at the end of the ten miles. If, however, it had exactly ten miles to travel and exactly twenty minutes to do it in, and must be at full stop at the beginning and at the end of the given time, it could not take up its full speed in an instant nor lose it in an instant, but must start gradually and stop gradually, and therefore during the first part and the last part of the time would be moving at a speed slower than $\frac{1}{2}$ mile per minute, and must go faster than $\frac{1}{2}$ mile per minute during the middle part of the run to make up for the time lost in starting or stopping, or, in other words, the motion would not be uniform. The more suddenly the train starts and stops, the more nearly uniform the motion may be, but the greater the shock when starting and stopping. Let us take another example on a smaller scale. Suppose we have some sort of a piece to be moved one foot across a machine in ten seconds. if the motion is to be uniform, the piece would have a velocity of $\frac{1}{10}$ foot per second. If the piece is light and the mechanism which does the moving is sufficiently powerful, the piece may be made to start and stop almost instantly and consequently may have what is practically uniform motion. It is evident, however, that the shock at starting and at stopping would be considerable.

Varying Motion. If the mechanism which moves the piece is so designed as to start and stop it gradually, the shock will be avoided. The character of this gradual increase and decrease of speed might be one of several kinds, but what would most likely be met in practice would be either a gradual increase of speed at the start, until full speed is attained, then a uniform full speed during the main part of the stroke, and finally a gradual decrease of speed to a full stop at the end of the stroke; or the speed would increase during the entire first half of the stroke, and decrease during the entire last half, the motion at no time being uniform. The first condition is most likely to be used where the piece is doing work during the stroke, as, for instance, the cutting stroke of a tool, and the second condition is used where it is desired to get the piece across quickly without regard to the character of the motion as long as it is without shock. The character of the motion usually employed in this case is either what is known as *harmonic motion* or *uniformly accelerated and retarded motion*, and we will now study these.

Harmonic Motion. In Fig. 28 let the semicircle A B D represent the path along which a piece moves with a uniform velocity. Now, if we have another piece moving along the diameter A D, starting from A at the same time that the first piece does, and moving at such a speed that a perpendicular let fall from any position of the first piece to the line A D will strike where the second piece is,—that is, when the first piece is at C the second piece is at E; when the first piece is at B the second piece is at O, and so on, — then the piece which travels across the line A D is said to have *harmonic* motion, and it is evident that it moves slowly at first, increases to a maximum speed at the center, and decreases towards the end.

Uniformly Accelerated and Retarded Motion. A piece which has uniformly accelerated and uniformly retarded motion travels at such a speed that it will move through one unit of space in the first unit of time, three the second, five the third, seven the fourth, nine the fifth, etc., to the middle of its stroke, then decreases at the same rate to the end of the stroke. For example, if a piece is to move with uniformly accelerated and retarded motion one foot in ten seconds, it will move $\frac{1}{50}$ foot the first second, $\frac{3}{50}$ the second, $\frac{5}{50}$ the third, $\frac{7}{50}$ the fourth, $\frac{9}{50}$ the fifth, when it will have traveled $\frac{1}{50} + \frac{3}{50} + \frac{5}{50} + \frac{7}{50} + \frac{9}{50} = \frac{25}{50}$ or $\frac{1}{2}$ the whole foot in five seconds, or $\frac{1}{2}$ the whole time, the speed increasing all the time; at the end of the fifth second, or in other words, when half the distance has been traveled, it begins to slow down, and travels $\frac{9}{50}$ the sixth second, $\frac{7}{50}$ the seventh, $\frac{5}{50}$ the eighth, $\frac{3}{50}$ the ninth, and $\frac{1}{50}$ the tenth. The rate at which the velocity increases during the first half of the time is the same as that at which the velocity of a weight will increase if dropped from a height, and the rate at which the velocity decreases during the last half of the time is the same as that at which the velocity of a ball will decrease if thrown straight up into the air. Uniformly accelerated or uniformly retarded motion is therefore often spoken of as *gravity* motion.

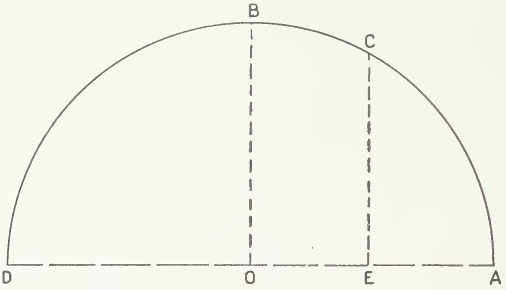


Fig. 28.

One of the most common means of obtaining the kinds of motion just described is by cams, although these are not often used for motions greater than a few inches.

Kinds of Cams. Cams may be divided into two general classes, *plate cams* and *cylindrical cams.* Either one of these may be designed for uniform motion, for harmonic motion, for gravity motion, or for a combination of two or even of all three.

Plate Cams. Fig. 29 shows the front and side views of a plate cam designed to raise the sliding piece P a distance A *e*; the lowest position of P being when its point is at A, and the highest position being when its point is at *e*, as indicated by the dotted lines. The cam turns right-handed, as shown by the arrow, and raises P with uniform motion from A to *e* while turning through $\frac{2}{3}$ of a revolution, then lowers P with uniform motion to its original position while turning through the remaining $\frac{1}{3}$ revolution.

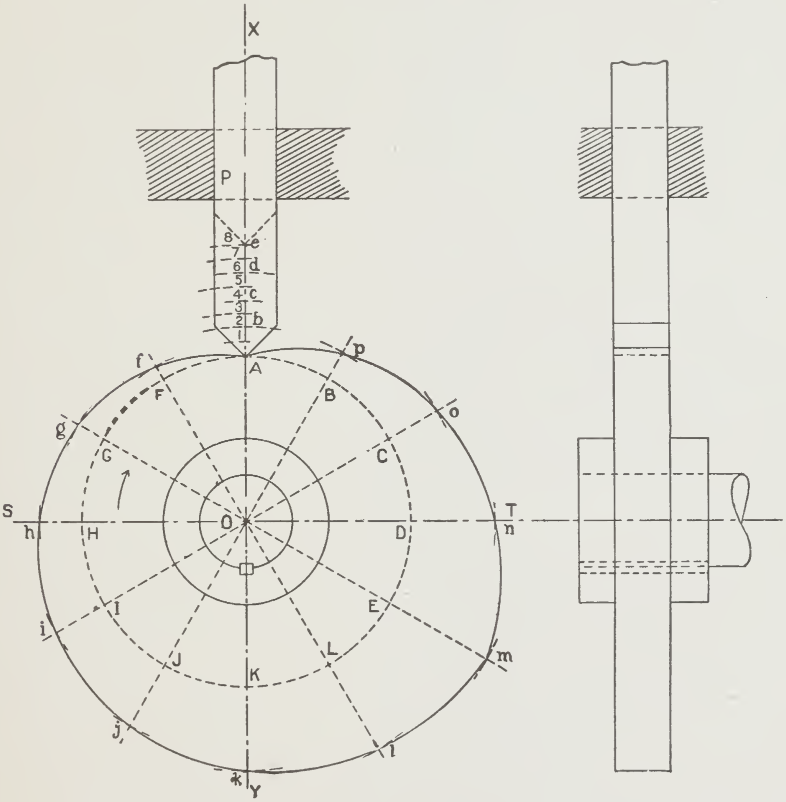


Fig. 29.

To design such a cam, start with the point O (the center of the shaft on which the cam is fixed); through O draw the center line X Y, and locate point A by measuring up from O the distance O A, which must be known; from A lay off A *e*, which must also be known; with O as a center and radius O A draw the circle A H K D; on this circle find a point E such that arc A H E shall be to arc A C E as the time allowed to raise P is to the time allowed to lower P. In this case, since P is to rise in $\frac{2}{3}$ of a revolution of the cam, and fall in $\frac{1}{3}$ of a revolution, the time allowed for it to rise is twice the time allowed for it to fall, and therefore the arc A H E is twice the arc A C E, or, in other words, the arc A H E is $\frac{2}{3}$ of the whole circumference, and A C E is $\frac{1}{3}$ of the circumference. If the cam were to turn left-handed instead of right-handed, the arc A C E would be $\frac{2}{3}$ of the circumference and A H E $\frac{1}{3}$. Now since the motion is to be uniform, divide line A *e* into any number of *equal* parts (eight are used in the figure) and divide the arc A H E into the same number of equal parts; draw the radial lines O F, O G, etc., as shown; with O as a center and radius O 1 draw an arc cutting O F at *f;* with O as a center and radius O 2 cut O G at *g*, and so on around to *m*. A smooth curve drawn through the points A, *f*, *g*, *h*, etc., around to *m*, will form the outline of that portion of the cam which raises P. Now it is evident that when the cam has turned through the angle A O F the line O F will coincide with the center line X Y, and the point *f* will coincide with point 1, and therefore the cam will have pushed the point of P from A to 1. In the same way, when the cam has turned through angle A O G, point *g* will coincide with 2, and when cam has made $\frac{2}{3}$ of a revolution *m* will coincide with 8 (which is the same as *e*), and P will have been raised the required amount.

The outline of the rest of the cam is found by dividing arc A C E into any number of equal parts, and making a new division of A *e* into the same number of equal parts (*ed*, *dc*, etc.). Points *n*, *o* and *p* are found in the same way as *f*, *g*, etc., and a smooth curve drawn through them.

Fig. 30 shows a plate cam similar to that shown in Fig. 29, except that in Fig. 30 the piece P is raised from A to *e* during $\frac{1}{2}$ a revolution of the cam, and lowered during the other $\frac{1}{2}$ revolution; that is, the time allowed for raising P is the same as the time allowed for lowering it. The motion is uniform in both raising and lowering. The method of drawing this cam is similar to that explained in Fig. 29, except that $\frac{1}{2}$ the circle A H

K D is divided into the same number of equal parts as the line A e instead of $\frac{2}{3}$ as before, k now being the point farthest from the center instead of m as in the previous case. This form of cam is called a *Heart Cam.*

Fig. 31 shows another cam, which is similar to that of Fig. 29. It will be noticed, however, that in Fig. 29 the line in which A travels coincides with the center line, X Y, or, in other words, the line of motion of point A passes through the center of the cam. In Fig. 31 the path of A does not pass through the center of the cam, but lies in the line V W, which passes a distance R to one side of the center O. The cam shown in the figure is designed to raise P a distance A e in $\frac{2}{3}$ of a revolution,

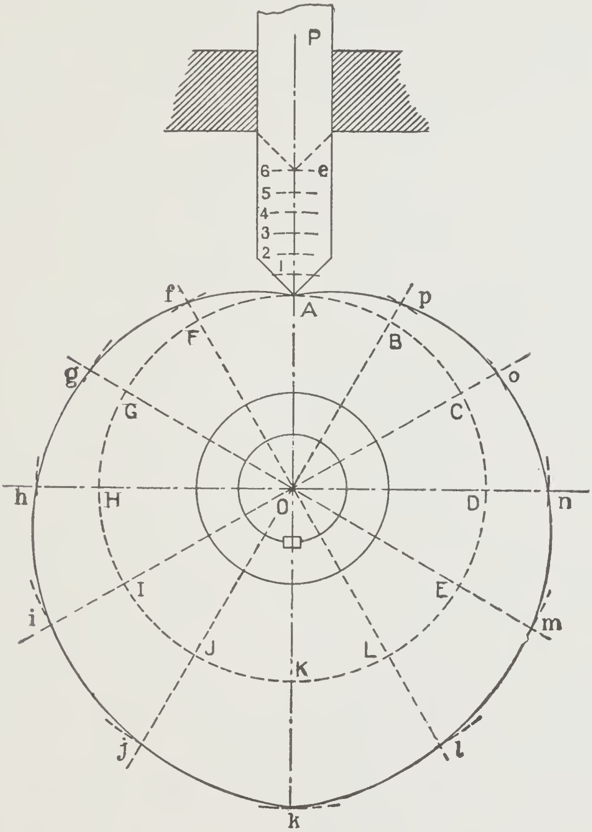


Fig. 30.

and lower it in the remaining $\frac{1}{3}$, just as in Fig. 29, the motion being uniform. To design the cam, locate center O and draw the center lines X Y and S T at right angles with each other. On S T measure from O a distance equal to R, which must be known, thus getting the point *a;* through *a* draw V W parallel

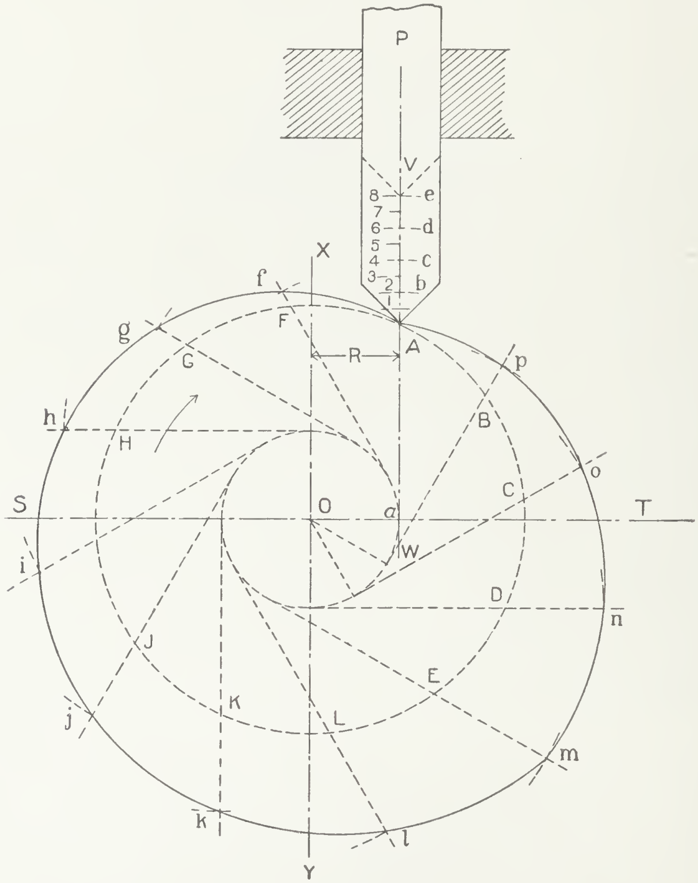


Fig. 31.

to X Y. Locate A and *e* on this line, the distances A *a* and A *e* being known. Through A draw a circle with O as a center; on this circle locate point E so that arc A H E shall be to arc A C E as time required to raise P is to time required to lower it; in this case as $\frac{2}{3}$ is to $\frac{1}{3}$, or, in other words, arc A H E is $\frac{2}{3}$ of the circumference. Divide A *e* into any number of equal parts, and divide arc A H E into the same number of equal parts. With O as a center and radius O *a*, draw a circle tangent to V W, and through the points of division F, G, H, etc.; on the large circle draw lines tangent to the smaller circle. With O as a center draw an arc through point 1, cutting line through F at *f;* points *g*, *h,* etc., are found in the same way by drawing arcs through 2, 3, etc. A smooth curve drawn through these points around to *m* gives the outline of that part of the cam which raises the point of P from A to *e*. The curve through *n*, *o*, *p*, which lowers P, is found in the same way. It will be noticed that the only difference between the method of laying out this cam and that in Fig. 29 is that the lines on which points *f*, *g*, *h*, etc., are found are drawn tangent to the small circle instead of to the center. The hub and bore for shaft are omitted on this drawing to avoid confusing the lines.

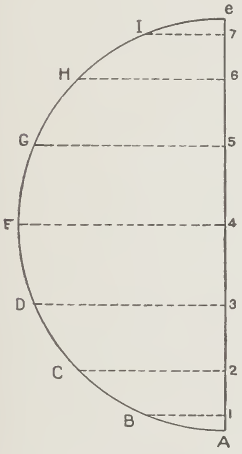


Fig. 32.

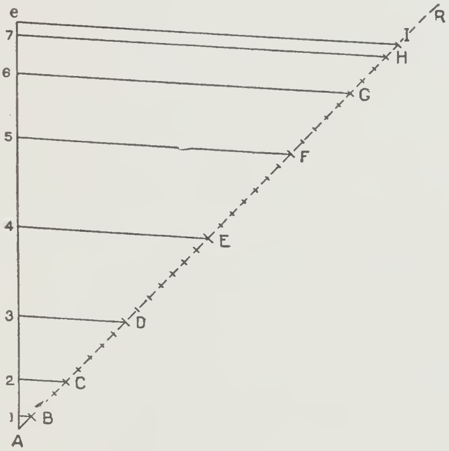


Fig. 33.

If, in Fig. 29, it had been required to raise and lower P with harmonic motion instead of uniform motion, the only difference in construction would have been in the dividing up of the line A *e*. The divisions A F, F G, etc., on the circle, would be the same as for uniform motion; that is, they would be equal to each other. The divisions A–1, 1–2, etc., on line A *e*, instead of being equal would be found as shown in Fig. 32. On the line A *e* as a diameter, draw a semicircle, and divide this semicircle into as many equal parts as we divide the arc A H E in Fig. 29. From the points B, C, D, etc. (Fig. 32), draw perpendiculars to line A *e*, cutting it at points 1, 2, 3, etc. These are the points which we would use for finding points *f*, *g*, *h*, etc., on the cam outline (Fig. 29). Fig. 32 is drawn to a larger scale than Fig. 29.

Fig. 33 shows how to divide the line A *e*, if the motion were to be uniformly accelerated and uniformly retarded. Since the line A *e* is to be divided into the same number of parts as the arc A H E, it will be divided into eight parts in this case. Now if the motion which the cam is to give to the piece P is to be uniformly accelerated and retarded, the line A *e* must be divided up in such a way that the distance from 1 to 2 is three times the distance from A to 1; the distance from 2 to 3 is five times the distance from A to 1; 3 to 4 is seven times A to 1; 4 to 5 is seven times A to 1; 5 to 6 is five times A to 1; 6 to 7 is three times A to 1; and 7 to *e* is equal to A to 1. Therefore the whole line A *e* is $1 + 3 + 5 + 7 + 7 + 5 + 3 + 1$, or thirty-two times the distance from A to 1, or, in other words, A–1 is $\frac{1}{32}$ of the whole line A *e*, 1–2 is $\frac{3}{32}$ of A *e*, and so on. To divide up A *e* so that the divisions may bear the above relation to each other, draw the line A R at any convenient angle, and choosing any convenient distance as a unit, mark it off on this line thirty-two times, beginning at A; from I, the last of these dividing points, draw a line to *e;* next find the points B, C, D, etc., as follows: B is the first division from A, C the third from B, D the fifth from C, E the seventh from D, F the seventh from E, G the fifth from F, H the third from G; through the points H, G, F, etc., draw lines parallel to I *e*, cutting line A *e* at 7, 6, 5, 4, etc.; then these points will be the required points of division for the line A *e*.

Fig. 34 shows a cam designed to raise P with uniformly accelerated and retarded motion, while the cam is turning right-handed through $\frac{1}{3}$ of a revolution; then P is to be held fixed in its raised position while the cam turns through the second $\frac{1}{3}$ revolution, and to be lowered with harmonic motion during the last $\frac{1}{3}$ revolution. The construction involves the methods already described. The arcs A F M, M N V and V E A are each $\frac{1}{3}$ of the circumference. The dividing of the line A *e* for uniformly accelerated and retarded motion is shown at the left of A *e*, giving points 1, 2, 3, etc., which are swung on to the lines O G, O H, etc., and thus locating points *g*, *h*, etc., as in the previous cams.

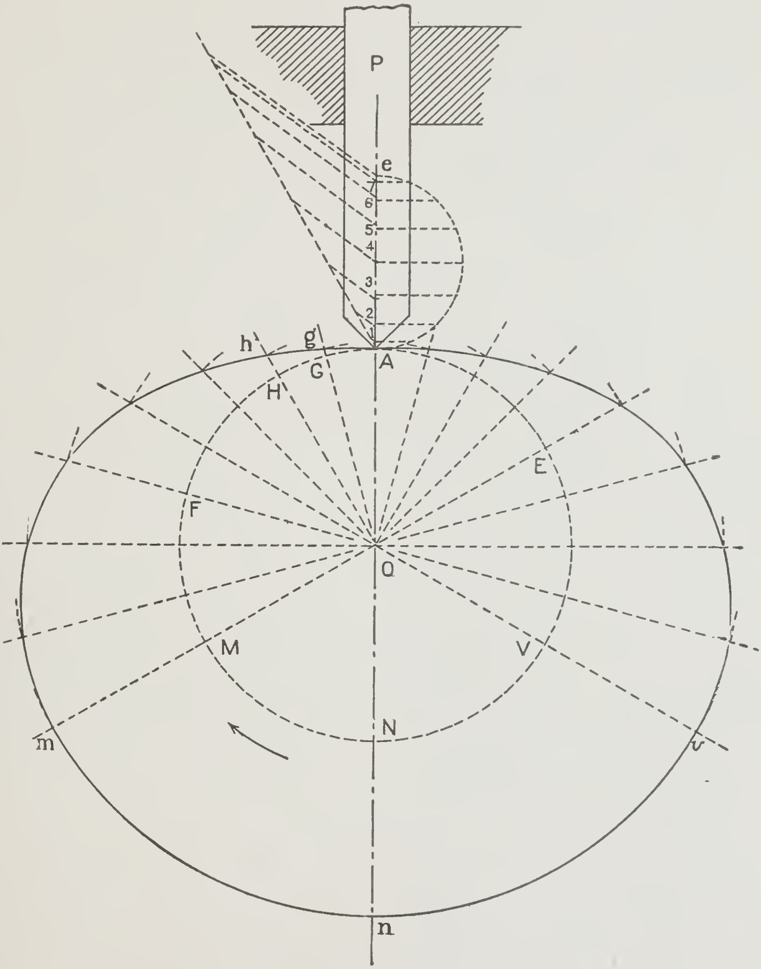


Fig. 34.

When the cam has turned through the angle A O M, the line O *m* coincides with line O *e*, point *m* being at *e*, and the point of P will be raised to *e*. Now the point of P is to remain at *e* while the cam turns through the next $\frac{1}{3}$ revolution, that is, while it turns through the angle M O V and therefore that part of the outline of the cam from *m* to *v* is an arc of a circle drawn with O as a center. The dividing of A *e* for the harmonic motion for lowering P is shown at the right of A *e*, and the points on the cam outline are found in the same way as before.

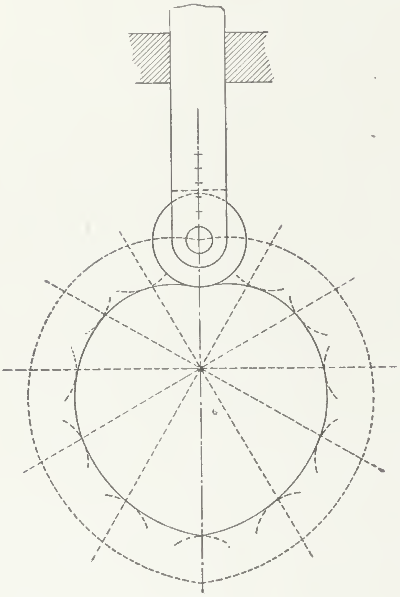


Fig. 35.

The cams which we have been considering all act on the comparatively sharp end of the piece P, and it is evident that, as the cam turns, its surface must slide on P, causing friction and rapid wearing. If, however, the end of P is provided with a roller which bears on the surface of the cam, the sliding and consequent wearing are avoided. Fig. 35 illustrates this, and shows how to lay out the cam. The cam here shown is to give the same kind and amount of motion to the follower P as the cam shown in Fig. 30. We first find the curve for the cam outline, if it acted on a pointed piece, as in Fig. 30, the center of the roller being at the place where the point of P is in Fig. 30. This curve is shown dotted in Fig. 35, and is exactly the same curve as the outline of the cam in Fig. 30. Now, with a radius equal to the radius of the roller which is to be used, and with centers at any number of points around the curve, draw arcs as shown. A smooth curve drawn *tangent* to these arcs will be the outline of the cam to act on the roller.

Cylindrical Cams.* A plate cam, as we have seen, can be designed to give almost any kind of motion in a plane at right angles to the axis of the shaft on which it is located, but cannot give motion in a line parallel to the axis of the shaft. If, however, we put a cylinder on the shaft with a groove cut in it, the

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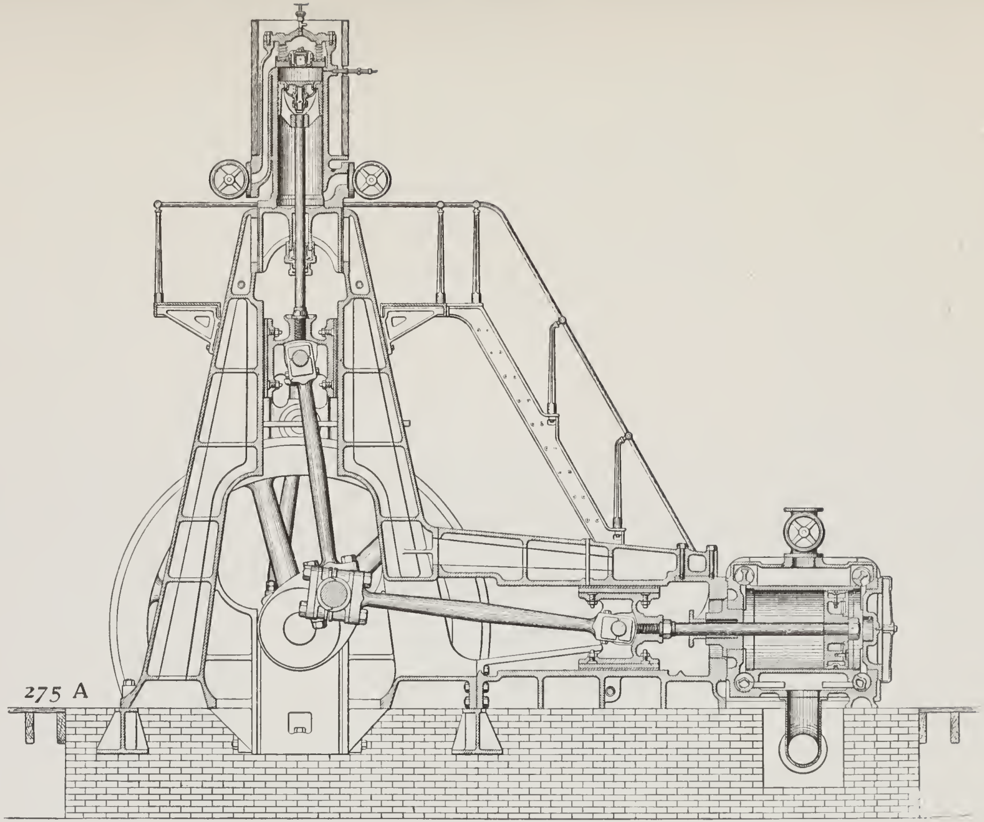
*The subject "Cylindrical Cams" is optional.



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shape of the groove may be made such as to give to a piece inserted in it almost any desired motion in a line parallel to the axis of the shaft. Such a cam is called a cylindrical cam, and is shown in Fig. 36, which, together with Figs. 37 and 38, shows the method of designing. This cam is designed to turn in the direction indicated by the arrow, and in so doing is to move a roller a distance A to the right with harmonic motion while turning $\frac{1}{4}$ revolution, to allow the roller to remain stationary during the next $\frac{1}{4}$ revolution, to move it back to its original position with uniformly accelerated and retarded motion during the third $\frac{1}{4}$ revolution, and to keep it stationary during the last $\frac{1}{4}$ revolution.

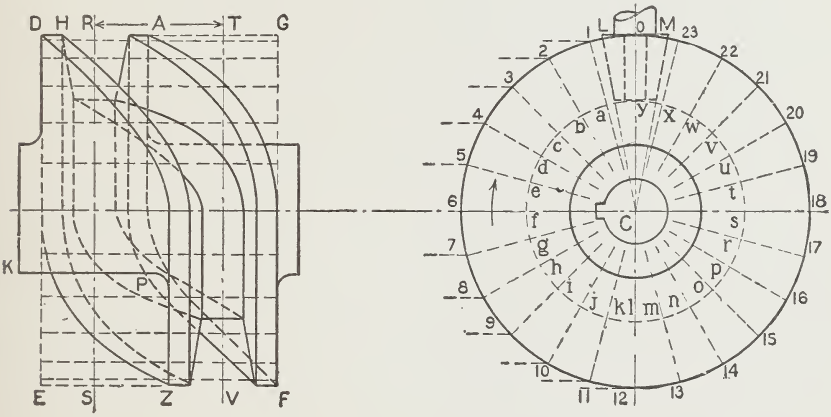


Fig. 36.

The roller is shown in the end view of the cam, and is of a diameter L M at its outer end, and of a length 0 y; it is made tapering, being the frustum of a cone whose apex is at the center of the cam. The depth of the groove is made exactly equal to the length of the roller, 0 y, and the width of the groove is made equal to the diameter of the roller. The arrangement might be a little different in practice, but this will show the method of designing.

The length of the cylinder which forms the cam is D G, which is equal to A (the distance between the two extreme positions of the center of the roller) plus twice the radius of the roller, plus twice D H (the thickness of metal allowed between the groove and the outside). In the figure the hub is allowed to project a little at each end.

To lay out the cam: Draw the two views of the cylinder, the side view being the rectangle D E F G, and the end view being the circle 0–6–18. We will start with the point 0. Since the roller is to move through distance A while the cam turns $\frac{1}{4}$ revolution, we lay off from 0 the arc 0–3–6 equal to $\frac{1}{4}$ the circumference; and the groove in this part of the cylinder must be such that when the cam has turned in the direction shown by the arrow until point 6 coincides with 0, the axis of the roller will have moved along in harmonic motion from a position coincident with center line R S (side view) to a position coincident with center line T V. The axis of the roller is to remain coincident with T V during the next $\frac{1}{4}$ revolution of the cam, that is, until point 12 comes to 0, and consequently the groove in this part of the cam goes straight around, the projection of its center line coinciding with T V. The roller is to be moved back until its axis coincides with R S with uniformly accelerated and retarded motion, while the next $\frac{1}{4}$ revolution is made, or until point 18 comes to 0, and the groove in this $\frac{1}{4}$ of the cam must be made such as will give that motion. The groove in the remaining $\frac{1}{4}$ of the circumference is again to be such as to keep the axis of the roller stationary, or, in other words, is to have the projection of its center line coincident with R S.

Now, in order to draw the projection of the groove in the side view, it will be necessary to make a development of the surface of the cylinder, using the same principles that were involved in the development of cylinders in Part III. Fig. 37 shows the development, and we will now consider in detail the method of making and using it. In Fig. 36 divide each of the arcs 0–6, 6–12, 12–18, 18–0 into any number of equal parts. In making the division it is more convenient to make the spaces into which these arcs are divided equal for all the arcs — that is, 1–2, etc. = 6–7, etc. = 12–13, etc. = 18–19, etc. On the side view of the cylinder (same figure) draw the elements corresponding to those points of division. Next draw the rectangle D E D′ G′ F G (Fig. 37), the length of the side D E D′ being equal to the circumference of the circle 0–6–18 (Fig. 36), and the length of the side G D being the same as the line G D in Fig. 36. On D E D′ locate points 6, 12, 18, by making distances D–6, 6–12, 12–18, 18–D′ equal respectively to arcs 0–6, 6–12, 12–18, 18–0, in Fig. 36. Divide each of the distances D–6, 6–12, etc., into the same number of equal parts as there are divisions in the corresponding arcs in Fig. 36, and at these points of division, which are numbered 1, 2, 3, etc., to correspond with the points in Fig. 36, erect perpendiculars as shown. On D G (Fig. 37) locate point *r*, making D *r* equal to the distance from D to line R S (Fig. 36), and draw line *r s* parallel to D D′.

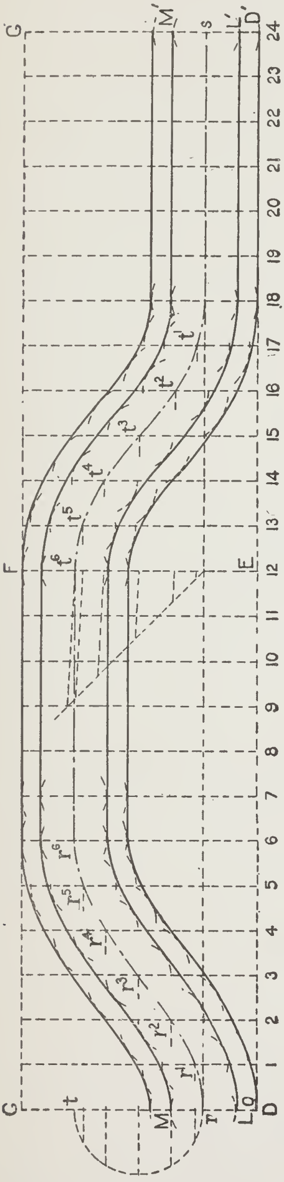


Fig. 37.

From *r* lay off *r t* equal to A, and divide *r t* for harmonic motion as shown, making the number of divisions the same as those between D and 6. From the first point of division project across to the perpendicular which was erected at 1, from the second point of division to the perpendicular which was erected at 2, etc., thus getting the points r^1, r^2, r^3, r^4, etc., to r^6. Through these points draw a smooth curve, as shown in dot and dash. The perpendicular distance from any point on this line, as r^3, to the line *r s*, represents the distance that the axis of the roller has moved to the right of the center line R S, while the cam has turned through the angle corresponding [through angle 0 C 3 (Fig. 36), if we consider the point r^3]. Since that part of the circumference of the cylinder corresponding to a length of arc equal to line 6–12 is to have a groove that will hold the roller stationary, the line r^6 t^6 is a straight line parallel to r s. Divide the distance from t^6 to r s for uniformly accelerated and retarded motion, and find points t^5, t^4, etc., in similar way as we found r^1, r^2, etc., and draw the curve through them. The line which we now have is the development of the center line of the groove. With a radius equal to the radius of the roller, and with centers at any number of points along the curve which has just been found, draw arcs as shown, and draw smooth curves tangent to these arcs, thus getting the curves L L′ and M M′, which are the developments of the sides of the groove. Now with the dividers take the distance from the line r s to where the curve M M′ intersects the line D G, and lay this distance off from line R S along line D G of Fig. 36; in like manner take the distance from r s to where curve M M′ intersects perpendicular at point 1, and lay this off from R S along the element which corresponds to point 1 in Fig. 36. Do the same for the points where M M′ intersects at 2, 3, etc., and also for points where L L′ intersects these perpendiculars; distances above r s being measured to the right of R S, and distances below r s to the left of R S. We thus obtain two series of points on the side view of the cylinder in Fig. 36, and smooth curves drawn through these points give the projections of the edges of the groove on the surface of the cylinder.

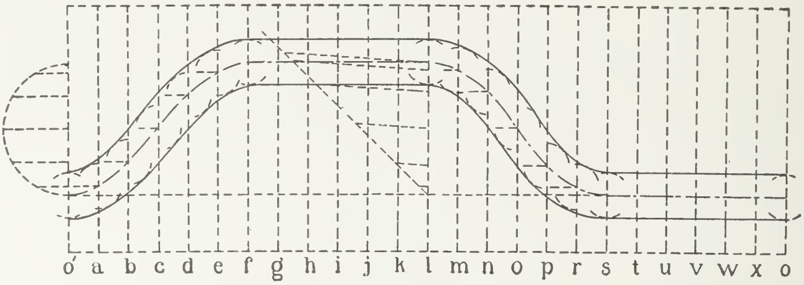


Fig. 38.

To get the projections of the bottom corners of the groove, we must make a similar development of a cylinder whose length is the same as the one which we have just developed, but whose diameter is the diameter of the larger one less twice the depth of the groove shown by the dotted circle in the end view of Fig. 36. The development is shown in Fig. 38, and is done in a manner exactly similar to that employed in Fig. 37. The method of getting the corresponding curves in the side view of Fig. 36 is also the same as for the curves on the outer cylinder.

We would now have a cam which would appear as in Fig. 39; but it is evident that there is a good deal of unnecessary metal, so that instead of making it a complete cylinder, the stock is usually cut away, as shown by the outer curves in Fig. 36, so that the distance from the center of the groove to the outside of the metal is everywhere equal to the distance from D to line R S. The projections of these outer edges of the metal are found by first making the development in Fig. 37, obtaining the upper and lower of the curves by striking arcs from the center curve with a radius equal to distance from D to R S (Fig. 36), and drawing the curves tangent to these arcs. The curves are transferred to the side view of Fig. 36 in the same way that the rest of the curves were. The hub is now drawn, extending as far beyond the rest of the cam as is desired, the contour of the hub (as K P) being visible to a point perpendicularly opposite where the outside edge of the cam strikes the contour of the cam (point Z for line K P).

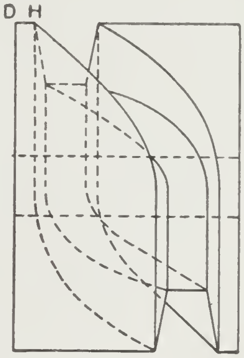


Fig. 39.

BELTING.

The word shaft, in connection with machinery, is a name applied to a bar, either hollow or solid, usually of circular crosssection (in common language, round), supported in suitable bearings so as to be free to turn, and intended to receive a motion of rotation from some source of power, and to transmit the motion to some other body, either another shaft or some piece of machinery. The number of times a shaft turns in a unit of time is called the speed of the shaft, and is usually stated as number of turns in a minute, or revolutions per minute, which is abbreviated to R. P. M.; this abbreviation will be used in what is to follow. When it is desired to drive one shaft from another at some distance from it, the connection is commonly made by placing on each shaft a pulley or wheel, and over these pulleys stretching a band or belt, as it is called, made of leather, rubber, canvas or some similar material. The pulleys are fastened to the shaft, so that a shaft and its pulley turn together, and the belt is stretched over the pulleys tight enough so that the friction between the belt and the surface of the pulleys is sufficient to make the linear speed of the belt practically the same as the linear speed of the circumference of the pulleys. The shafts which are connected by a belt may be parallel or nonparallel.

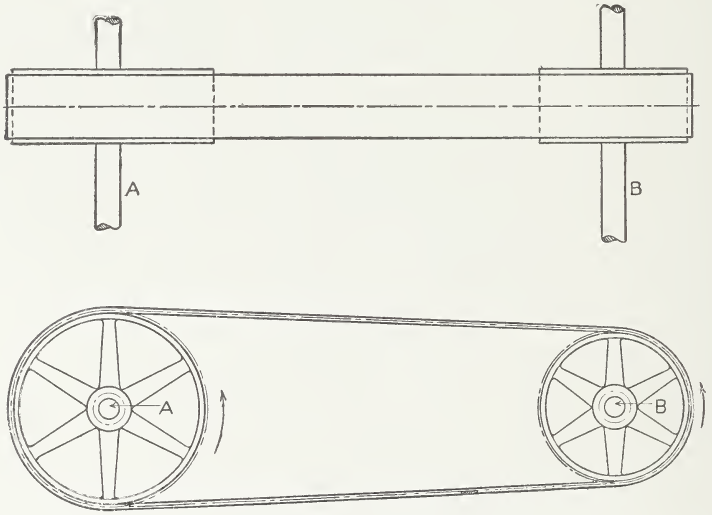


Fig. 40.

The methods of laying out the locations of pulleys and belts for the various cases which are commonly met with will now be considered. Before doing this, however, it will be necessary to consider a few general definitions and a few of the principles which govern belting.

Fig. 40 is an elevation and a plan of two shafts, A and B, which are connected by a belt. The *pitch line* of the belt is an imaginary line exactly in the center of the belt, both as regards thickness and width. The pitch line of each pulley is a circle drawn around the center of the pulley, with a radius equal to the radius of the pulley plus one-half the thickness of the belt. The pitch line of the pulley, therefore, coincides with the pitch line of that portion of the belt which is in contact with the pulley. The pitch lines of both the belt and the pulleys are shown in the figure by dot and dash lines. The diameter of this pitch circle of the pulley is called the effective diameter of the pulley, and should be used in place of the real diameter in all calculations and drawings for the location of pulleys, if such calculations and drawings are required to be absolutely accurate. The thickness of the belt, however, is almost always small compared with the diameter of the pulley, so that the error introduced by neglecting to make allowance for this thickness is so small as to be almost unnoticeable. It is generally safe to use the real diameter of the pulley instead of the diameter of its pitch circle, and this will be done in our discussion of the subject.

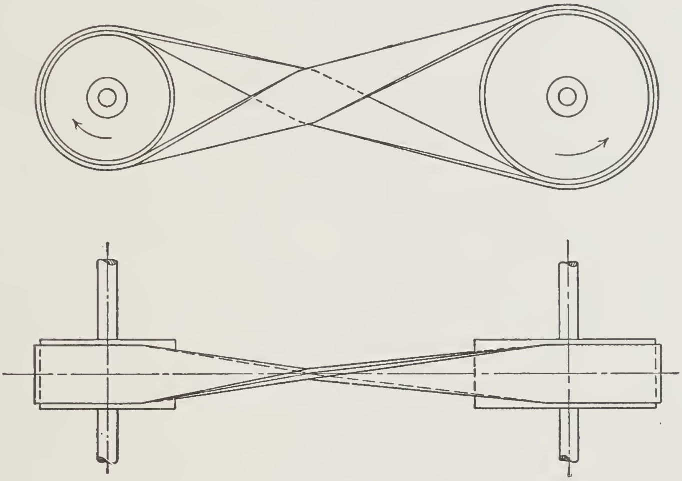


Fig. 41.

Parallel Shafts. *Open and Crossed Belts.* The belt shown in Fig. 40 is what is known as an open belt, and two shafts connected by such a belt will both revolve in the same direction. Fig. 41 shows a crossed belt, which causes the driven shaft to turn in the opposite direction from the driving shaft. The shafts in Fig. 40 and Fig. 41 are parallel to each other.

Shafts not Parallel. It very often happens that two shafts which are not parallel are to be connected by a belt, and this can always be done, whatever the position of the shafts with relation to each other, provided the pulleys are so located as to conform to one fundamental principle which governs the running of all belts. This principle may be stated as follows: The point where the center line of a belt leaves a pulley must lie in the plane which contains the pitch line of the pulley which receives the belt. Another way of stating this principle is: The center of a belt must be delivered into the center plane of the pulley to which it is running. This law is of great importance, as it applies to belts however they are bent or twisted, and the student should understand it clearly and have it fixed firmly in mind. From the study of geometry we learn that a plane is a perfectly flat surface, without thickness, and is supposed to extend indefinitely. The best example we can get of a plane is a thin, perfectly flat sheet of paper. The size of the paper may be increased indefinitely in all directions, but as long as the paper remains perfectly flat any two lines drawn on it will be in the same plane. So when we say that a certain point A on the surface of a pulley lies in the center plane of another pulley, we mean that if an imaginary flat surface is passed through the center of the second pulley, perpendicular to the shaft, and the flat surface is extended far enough, it will pass through the point A on the first pulley. This will be seen by referring to Fig. 42, which is a plan and two elevations of a belt connecting two shafts at right angles to each other, one shaft being above the other. The pulleys are arranged to turn in the directions indicated by the arrows. In the left-hand elevation it will be seen that the center line of that part of the belt which is moving downward leaves the upper pulley at the point A. The line X Y is the edge view (or trace) of the plane which passes through the center of the lower pulley, and in order that the belt may stay on the lower pulley, the lower pulley must be so located on its shaft that this line X Y shall contain point A. It will be noticed that the part of the belt which is moving upwards is drawn off the lower pulley at a considerable angle sidewise; but this is allowable, for a belt leaving a pulley may be drawn out of the center plane of the pulley. Now, referring to the right-hand elevation, we see that the upward-moving part of the belt leaves the lower pulley at B; S T is the trace of the center plane of the upper pulley, and this pulley must be so located on its shaft that S T shall contain the point B.

The detail for making the drawing for this kind of a belt will be considered a little later, under the subject of quarter-turn belts, the brief explanation at this point being given to make clear what is meant by the principle of delivering a belt into the plane of the receiving pulley.

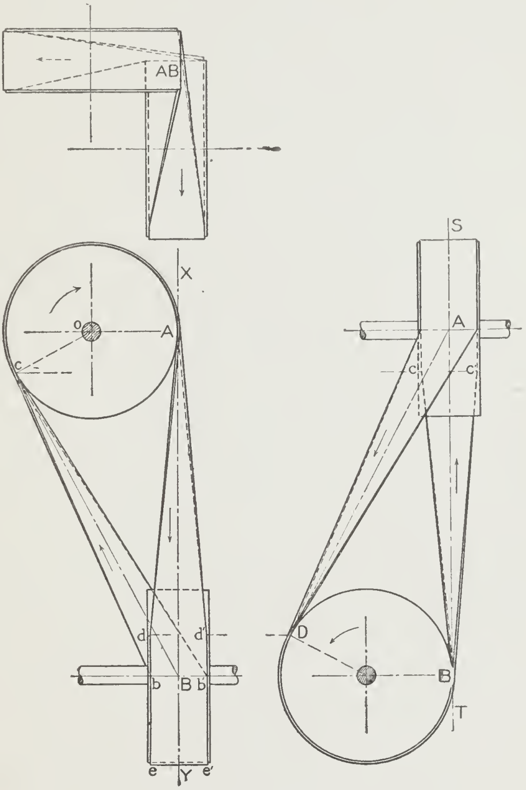


Fig. 42.

Quarter-turn Belt. A belt arranged to drive two shafts which are at right angles with each other, as shown in Fig. 42, is called a quarter-turn belt. There are various other methods of arranging the belt besides the one shown in Fig. 42, but as this is the simplest and easiest to understand we will study it first. With this arrangement the shafts must always turn in the same direction, for if the direction of rotation is reversed, neither pulley will deliver the belt into the center plane of the other pulley, and consequently the belt will not remain on the pulleys.

To draw the pulleys in their proper locations, having given the location of the shafts with relation to each other, the direction in which each shaft is to turn, and the sizes of the two pulleys, proceed as follows: Draw first the two views of each shaft at the given distance apart, and put on the arrows to indicate the direction of rotation. Start with either pulley, let us say the upper one, and draw that view in which it appears as a circle, which in this case will be the left-hand view. Draw line X Y tangent to the circumference of this pulley at point A on the side which is turning downwards, A being on a horizontal line drawn through the center of the shaft. The point where X Y intersects the center line of the lower shaft will be the center of the other pulley, which will appear in this view as a rectangle of a length equal to the diameter of the pulley, and width equal to the width of the face of the pulley. Now draw the other view of the lower pulley, which will be a circle, and draw the line S T tangent to the circumference of this pulley at the point B, on the side which is turning upwards. The point of intersection of S T and the center line of the upper shaft will be the center of the rectangle which represents the side view of the upper pulley. The plan view may be placed above either one of the elevations, and is made according to the ordinary principles of projections, by studying out from the two elevations already made how the pulleys are located with relation to each other, and drawing them as they would appear if looked at from above.

A careful study of the figure will show that point A is vertically over point B, so that in plan the projections of the two points coincide, or, as it is stated in shop language, the leaving sides of the pulleys are in the same plumb line.

We now have the drawing completed as far as is necessary to determine the location of the pulleys, it being unnecessary to draw the arms, hubs, etc., of the pulleys in a drawing which is made to show location of pulleys on the shafts. It usually makes the drawing clearer, however, if the belt is shown also, and to make the drawing of the belt, the width and thickness must of course be known. The belt should always be put on so that the same side of the belt touches both pulleys when it is possible to do so. We will call the side of the belt which touches the pulleys the inside, and the other side the outside. Now referring first to the left-hand elevation in Fig. 42, where the belt lies around the circumference of the upper pulley, from the point C, where the upward-moving part of the belt strikes the pulley, around to A, where the downward-moving part leaves the pulley, only the edge of the belt is visible, and is represented by drawing an arc of a circle from line O C around to O A with a radius equal to the radius of the pulley plus the thickness of the belt. The location of C may be found near enough for all practical purposes by drawing a line from B tangent to the upper pulley, C being the point of tangency, and O C is the radius drawn through C. In drawing the arc of the circle from O C to O A it is well to let it run by these lines a little way in the pencil drawing, and that part which is not used may be erased after the drawing is inked. The descending part of the belt leaves the pulley at A, and from that point to the place where it strikes the lower pulley it twists through an angle of 90°, coming out over the front of the lower pulley. The edge of the belt, which we see in its full thickness where it leaves the upper pulley at A, twists toward the left, less and less of it being seen as the belt descends, until at *d*, where it strikes the other pulley the two corners of this edge coincide, and from there to the bottom of the pulley we see this edge of the belt as a line. At the same time that the thickness of the belt has been disappearing from view, the outside of the belt has been coming into view, until at *d d′* we see the full width of the belt, the outside corner being the one which is visible. The inside corner is behind the rest of the belt and is shown dotted in the figure, although it is usually omitted entirely. From *d′* to the bottom of the pulley the two right-hand corners coincide. Points *d* and *d′* are found by drawing the horizontal dotted line through D (right-hand view) and laying off on this line a distance equal to one-half the width of the belt each side of line X Y. From *e e′* the belt goes up under and around the back side of the pulley to *b b′*, where it begins to be drawn off to the left, *b b′* being on the center line drawn through the shaft, since, by glancing at the right-hand view, we can see that the upward-moving part of the belt leaves the lower pulley at B, which is on a level with the center of the shaft. After leaving *b b′* the belt remains behind the pulley for a short distance, and is shown

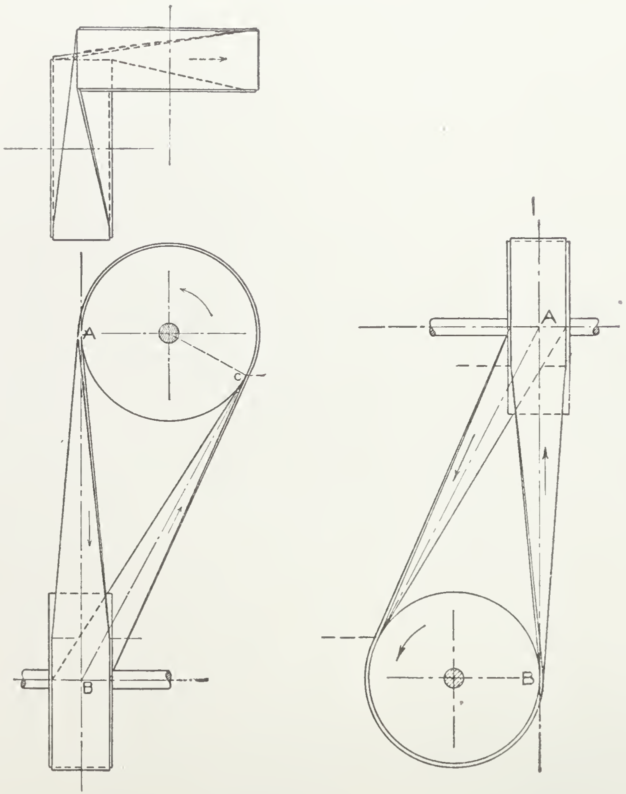


Fig. 43.

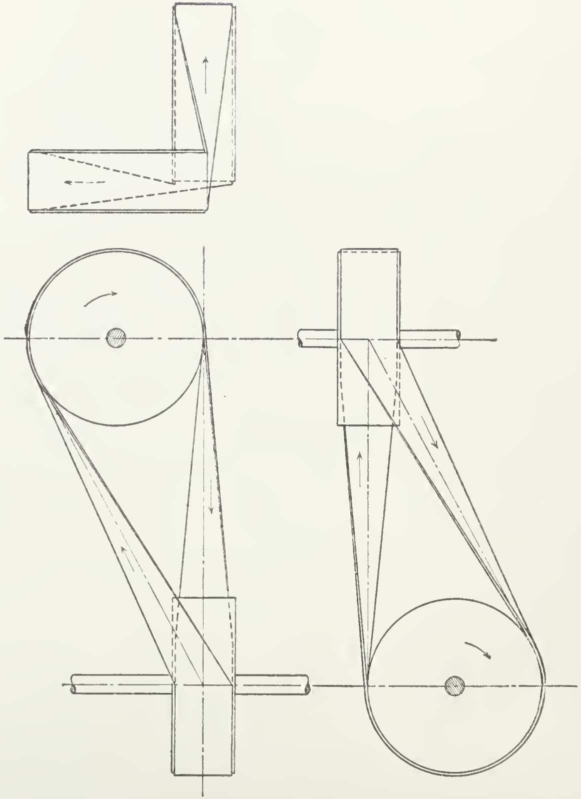


Fig. 44.

dotted. It is projected at its full width at b b', but as soon as it leaves the pulley it begins to twist toward the right, the left-hand edge gradually coming into view until it is seen at its full thickness where it strikes the pulley at C. At the same time, the *inside* face of the belt, which is toward the front when the belt leaves the pulley, gradually turns toward the right and appears narrower until it disappears from view, that is, is projected as a line. This completes the left-hand view, and the right-hand view is drawn according to the same principles, as is also the plan view.

An actual belt would not twist uniformly, and its corners would not appear as straight lines, but would curve, the form of the curve depending on the stiffness of the belt, the size of the pulleys, etc., but the straight lines are much easier to draw and represent the belt sufficiently well for all practical purposes.

Figs. 43 and 44 show what changes are made in the location of the pulleys and the appearance of the belt by changing the direction of rotation of the shafts. In Fig. 43 the lower shaft turns in the same direction as in Fig. 42, but the upper shaft turns in the opposite direction. In Fig. 44 the upper shaft turns in the same direction as in Fig. 42, and the lower shaft turns in the opposite direction.

Reversible Quarter Turn with Two Guide Pulleys. The quarter-turn belt of the kind shown in Figs. 42–44 works satisfactorily, provided there is never any reversing of the direction of rotation. Sometimes, however, it happens that the shafting in a shop or mill is caused to make a few turns in the opposite direction from that in which it ordinarily runs. This might be done accidentally, or intentionally in the course of making repairs, and, as has already been stated, if it should occur, a quarter-turn belt such as described above would immediately leave the pulleys. With a small belt this might not be a serious matter, as it could readily be replaced, but with a large belt it would cause delay and might cause damage. Accordingly, if a large quarter-turn belt is to be used, it is desirable to have it arranged so that it can travel in either direction if occasion required. To accomplish this, one or more guide pulleys are used in addition to the pulleys on the main shafts, these guide pulleys being placed on auxiliary shafts so located as to conform to the general principle of belting, that the belt may always be delivered into the plane of the pulley which is to receive it. When guide pulleys are used, the main pulleys should be so located with relation to each other that the *receiving* side of the *driving* pulley shall be in line with the *delivering* side of the *driven* pulley, so that the part of the belt which transmits the power from the driving to the driven pulley (called the tight side of the belt) may go direct without passing over the guide pulleys, thus allowing the latter to guide the loose side of the belt back from the delivering side of the driving pulley to the receiving side of the driven pulley.

We will first study the arrangement when two guide pulleys are used. Let the shaft R, Fig. 45, be the driving shaft, and P be the driven shaft, and let the ordinary direction of rotation of each be as indicated by the arrows. Then A is the point on the driving pulley which receives the center of the belt from the delivering point B of the driven pulley, and if R and P are horizontal shafts, A and B should be in the same plumb line, thus giving the tight part of the belt a direct passage from driven to driver, making a quarter turn in its passage. To draw the pulleys so that this condition may exist, start with either shaft (say R) and draw that view which will show the pulley on it as a circle (in this case the left-hand view). Draw the line X Y tangent to the circumference at A, and the point where X Y intersects the center line of the shaft P will be the center of the driven pulley, which will appear as a rectangle in this view. Next draw the other view of the driven pulley, which will appear as a circle, and at the point B on the delivering side draw the line X′ Y′ tangent to the circumference of the pulley. The intersection of this line and the center line of the shaft R will be the center of the side view of the driving pulley (being the other projection of the point A). It will thus appear that both views of the line A B are perpendicular to a horizontal line, or, in other words, A B is a plumb line. This gives the required locations of the two main pulleys relative to each other.

Next draw the plan view of these two pulleys, these being constructed from the two elevations, according to the ordinary principles of projection.

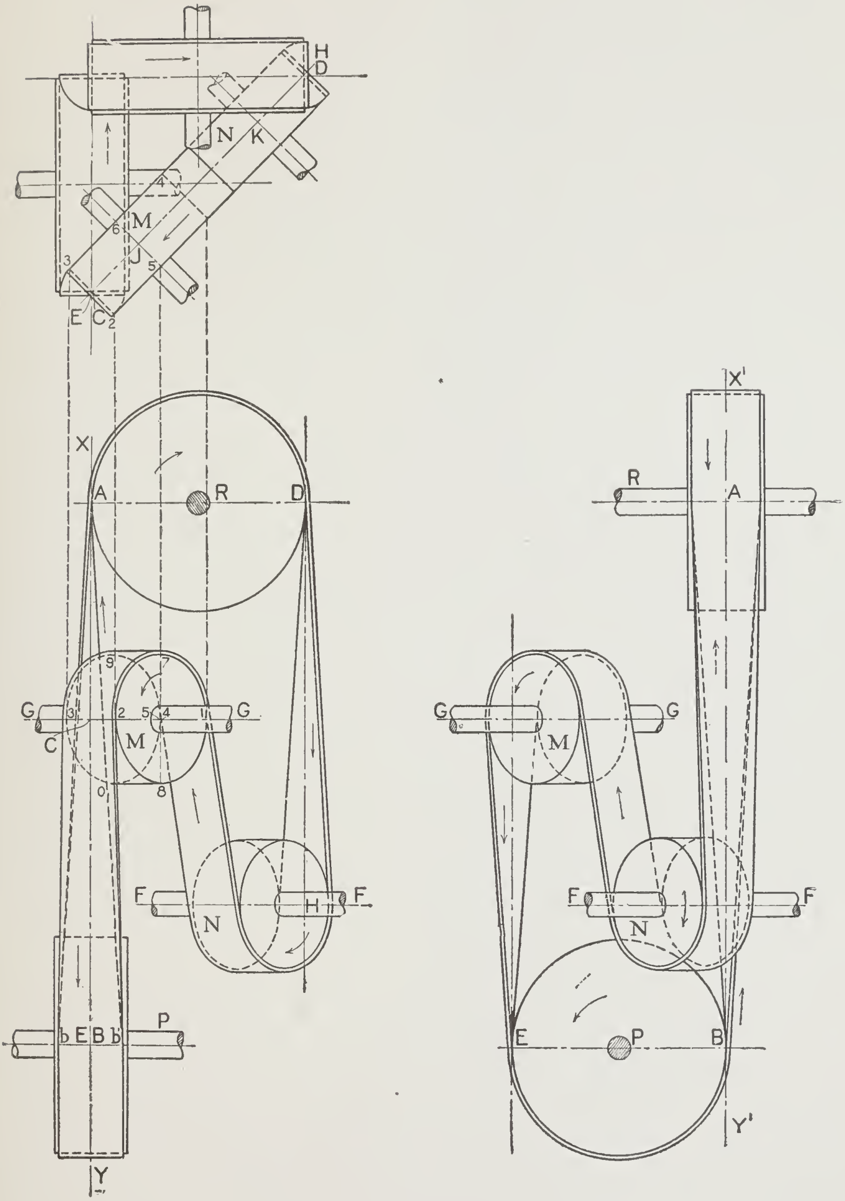
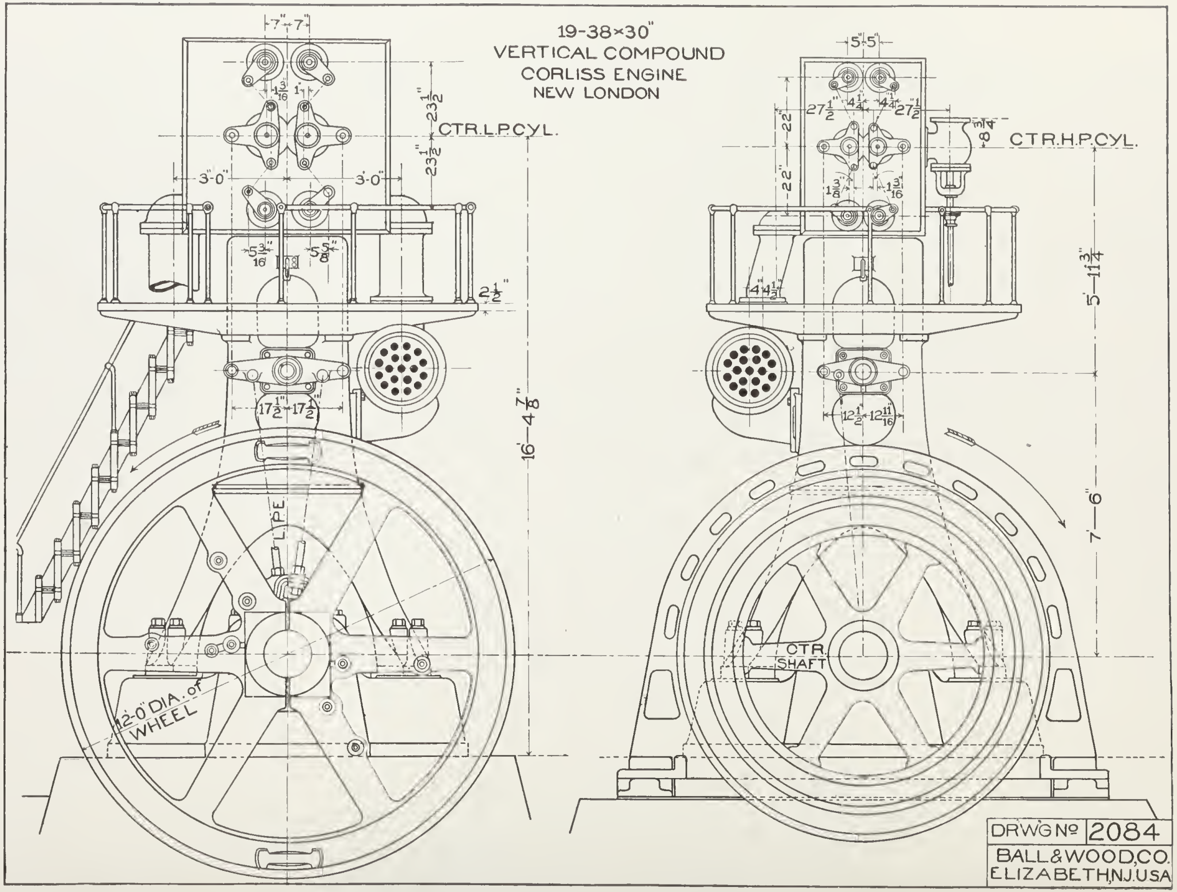

H
D
N
K
4
M
6
3
J
5
E
C
2
X
A
R
D
9
7
G
3
2
5
4
G
C
M
0
8
F
N
H
F
b
E
B
b
P
Y
X'
R
A
G
M
G
F
N
F
E
P
B
Y'

Fig. 45.

When two guide pulleys are to be used, the shafts on which both are located are horizontal if the main shafts are horizontal; the guide pulleys will therefore appear in plan as rectangles. We will accordingly draw the plan view of the guide pulleys first. They are to be so located that their center planes shall coincide, and that the point D, where the belt leaves the driving pulley, shall lie in the center plane of the guide pulley N, and that the point C, where the belt leaves the guide pulley M, shall lie in the center plane of the main driven pulley, in order that the belt will run in the direction indicated by the arrows. In order that the belt may run in the opposite direction if desired, the point E on the main driven pulley must lie in the center plane of the guide pulley M, and the point H of guide pulley N must lie in the center plane of the main driving pulley. In other words, points C and E must be in the same plumb line, and points D and H must be in the same plumb line. Then to draw the pulleys in plan, so that these conditions may exist, draw a line through E and D; this line will be the center plane of the two guide pulleys. From E, measure along the line E D a distance equal to one-half the diameter of the guide pulley M, and from the point J thus found draw a line perpendicular to the line E D; this line is the center line of the shaft on which M is located. In the same way, from D measure off along E D a distance equal to one-half the diameter of the guide pulley N, and draw a line perpendicular to E D through the point K thus found, giving the center line of the shaft on which N is located. Draw the rectangles as shown, representing the plan view of the guide pulleys.

To draw the elevations we must know the distance from the centers of the guide pulley shafts to the main shafts. This may be any convenient distance, it being desirable, however, to remove each guide pulley some distance from the main pulley which delivers to or receives from it, in order that the belt may not be obliged to turn in too short a distance. Draw the center line G G of the shaft for M, and F F of the shaft for N. The guide pulleys will appear in both elevations as ellipses. Consider first the pulley M in the left-hand elevation. Let fall the perpendicular (shown dotted) from point 5 in the plan, to center line G G, giving point 5 in the elevation. Along this perpendicular lay off



DRAWING OF VERTICAL COMPOUND CORLISS ENGINE.

Reproduced from a blue print.

the distances 5 – 7 and 5 – 8 each equal to one-half the diameter of pulley M. The line 7 – 8 is the major axis of the ellipse which represents the right-hand edge of M. Now from points 1 and 2 in plan, drop perpendiculars cutting G G at 1 and 2 in elevation. Line 1 – 2 is the minor axis of the ellipse, and the ellipse 1 – 7 – 2 – 8 may be drawn by any geometric method for constructing an ellipse when its two axes are known. The ellipse 4 – 9 – 3 – 0 is drawn in the same way, and the lines 9 – 7 and 0 – 8 drawn tangent to the two ellipses complete this elevation of M. The elevation of N is found in exactly the same way. The same principles are used in getting the other elevation of the guide pulleys.

We have now completed the drawings of all the pulleys, and it only remains to draw the belt as it would appear in the different views. Here, as in the preceding cases, we will represent the edges of the belt by straight lines in the elevations, although they would really be curves. In the plan, where the belt turns as it drops directly from one pulley to another, the edges are represented by arcs of circles. Let us start with the left-hand elevation, with the tight part of the belt (that is, the part which goes directly from B to A). As we are looking at the pulleys in this view, B is on the back side of the pulley, and at B the belt is projected at its true width; as it rises it twists toward the right, so that the edge *b'*, which was at the right when leaving the lower pulley, recedes to the back edge of the upper pulley, while the edge *b*, which was at the left when leaving the lower pulley, comes forward to the front edge of the upper pulley; and we see this edge of the belt in its true thickness where it comes tangent to the upper pulley, and all the way around this pulley until it leaves it again at D. Here it begins to twist to the right again as it goes to the guide pulley N, goes around the lower side of N, then up over and down on the front of M, leaves M opposite the center, and drops direct to the front side of the main pulley, twisting again to the right, completing the quarter twist that was partly made when dropping from D to H. The appearance of the belt in the right-hand elevation is reasoned out in the same way, and a careful study of the figure should make it perfectly clear how to draw the belt. It will also appear that each side of the belt comes successively in contact with the pulleys, so that the belt is put

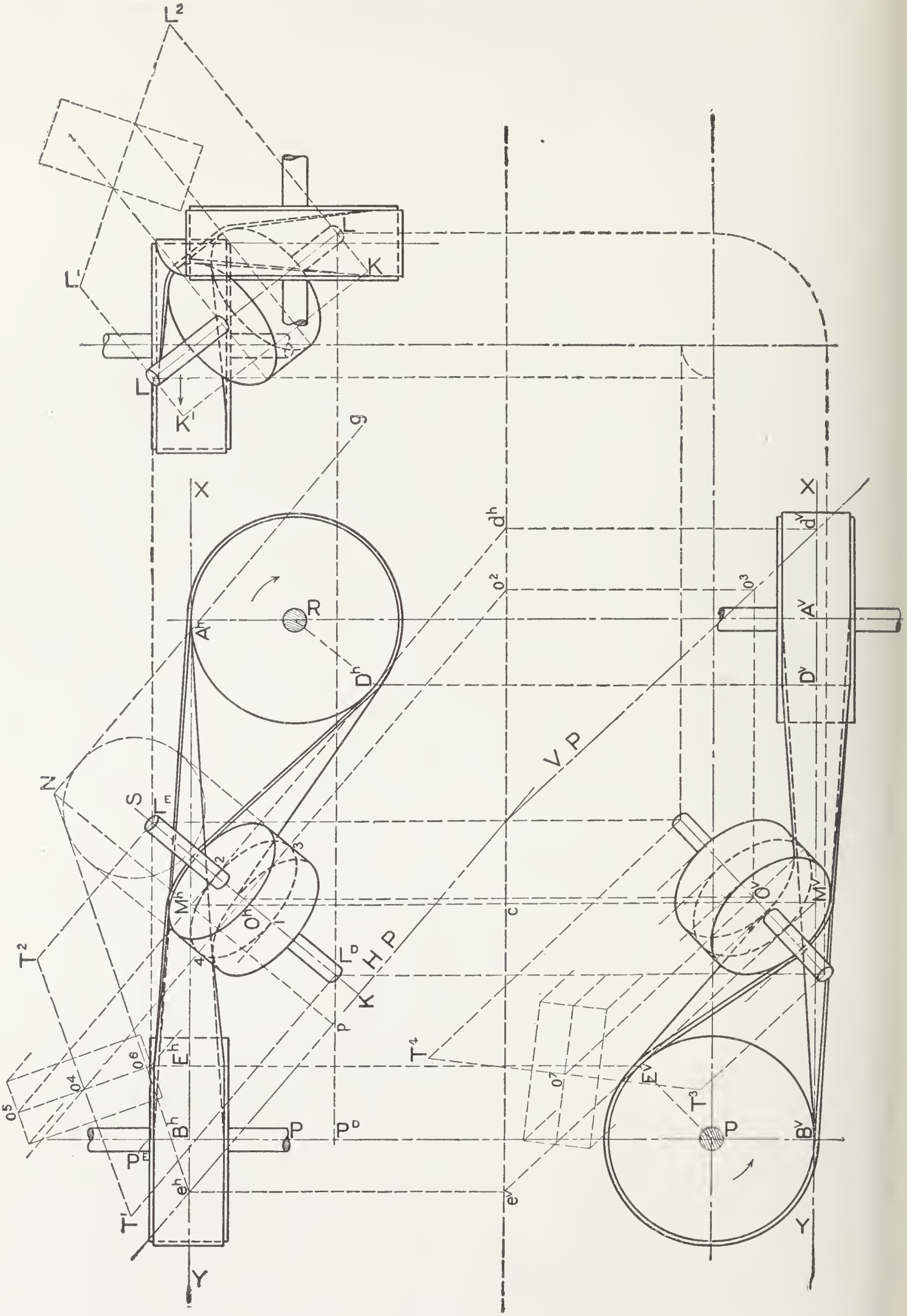

L^2
L'
L
K
L
K'
g
X
d^h
o^2
R
A^h
D^h
V P
N
S
L^E
2
3
M^h
O^h
1
4
L^D
H P
K
T^2
p
E^h
o^6
o^4
o^5
B^h
P
P^D
P^E
e^h
T'
Y
c
e^v
T^4
o^7
E^v
T^3
P
B^v
Y
X
d^v
o^3
A^v
D^v
O^v
M^v

Fig. 46.

together with a twist in it. It is usually desirable to have only one side the working side, but if such were the case with this arrangement of pulleys it would bring a sharp twist in the belt at some part, which might be a greater objection than having the belt work on both sides.

Reversible Quarter Turn with One Guide Pulley.* Fig. 46 shows the arrangement of the pulleys for the belt to run in either direction, using only one guide pulley. The ordinary direction of rotation is that shown by the arrows. The upper pulley is the driver, and, as in the case where two guide pulleys are used, the tight part of the belt goes directly from one main pulley to the other, the slack part of the belt returning over the guide pulley. The shaft of the guide pulley must be set at an angle with both main shafts in order to guide the belt properly. The method of locating the main pulleys is exactly the same as in the case where the two guide pulleys are used, so that the explanation given for that case will apply here, and all we need to consider in this case is the location and drawing of the guide pulley. We will first consider the two elevations. The plumb line X Y, as well as being the center line of the tight part of the belt, is the line of intersection of the center planes of the two main pulleys. Choose a point in the line X Y, which may be anywhere along the line, depending on how far the guide pulley is to be from one or the other of the main shafts, but preferably about halfway between them. The point is marked M^h in the left-hand elevation, and M^v in the right-hand elevation. From M^h draw a line tangent to the upper pulley at D^h, and from M^v draw tangent to the lower pulley at E^v. The other projection of the line $M^h D^h$ will be $M^v D^v$, coinciding with X Y in the right-hand elevation; and the other projection of $M^v E^v$ will be $M^h E^h$, coinciding with X Y in the left-hand elevation. We now have two lines, M D and M E (shown respectively by their two projections, $M^v D^v$ — $M^h D^h$ and $M^v E^v$ — $M^h E^h$), which determine the plane of the guide pulleys and which are practically the center lines of that part of the belt which passes over the guide pulley, and our problem is one of projections. The problem is to find on the drawing paper the traces of the plane which contains the two lines, and which is the center plane of the guide pulley. Revolve this plane about one of its traces until it is parallel to the plane of the paper, so that the true angle between the lines will be shown; then draw the guide pulley tangent to the lines in their revolved position, and revolve the lines back to their former position, revolving the guide pulley back at the same time. To carry out this construction proceed as follows: draw a ground line anywhere between the two elevations, parallel to X Y, and for the time being consider one of the elevations as a horizontal projection and the other as a vertical projection, remembering that our drawing is made as if projected on two planes located as in Fig. 5, Part IV, or, as it is commonly expressed, "in the third quadrant." We will treat the left-hand elevation as if it were the horizontal projection, and the right-hand elevation as if it were the vertical projection. Extend line $M^h D^h$ until it meets the ground line at d^h, and at d^h draw a perpendicular to the ground line meeting X Y (which is the same as $M^v d^v$ extended) at d^v. Through d^v draw a line parallel to $M^v E^v$, and this line, which is marked V P, is the vertical trace of the plane which contains the lines M D and M E. In like manner find the horizontal trace by extending $M^v E^v$ to meet the ground line at e^v, erecting a perpendicular at e^v to meet $M^h E^h$ at e^h, and drawing H P through e^h parallel to $M^h D^h$. If the work is correctly done, H P, V P and the ground line will intersect in a common point. Now through M^h draw a line perpendicular to H P, meeting H P at p. Construct the right triangle $t\ m^h$ P (Fig. 47), making m^h P equal to $M^h p$ on Fig. 46, and making $m^h t$ equal to the perpendicular distance of M^v from the ground line in Fig. 46 (that is, equal to $c M^v$). Then take the distance $p\ t$ (Fig. 47) and lay it off on the line $p\ M^h$ (Fig. 46) from p, thus obtaining point N. Join N and e^h, and through N draw N g parallel to $M^h D^h$. The lines N e^h and N g are the projections on the horizontal plane of lines M E and M D respectively when the plane P, which contains these two lines, is revolved so that it is parallel to the horizontal plane. Therefore the angle g N e^h is the true size of the angle between lines M E and M D. Now with a radius equal to the radius of the guide pulley which is to be used, draw a circle which shall be tangent to the lines N e^h and N g. This circle is the center circle of the guide pulley revolved parallel to the horizontal plane, and its center S is the revolved position of the center point of the guide pulley, and of course lies in the plane P. To revolve the center circle of the guide pulley back so as to get its two projections when it is in the position which it actually occupies with relation to the two main pulleys, we will first revolve back the point S to O^h. To do this, draw S K perpendicular to H P. Then in Fig. 47 lay off from P along the line P *t* the distance P V, equal to S K in Fig. 46. Care must be taken to lay off this distance from P rather than from *t*, and in order to remember from which point to measure, the student can bear in mind that distances measured along the hypothenuse from P (Fig. 47) represent distances measured from H P (Fig. 46). Having thus found point V, draw a line perpendicular to m^h P, meeting it in o^h, take distance o^h P in the dividers and lay it off from K along K S (Fig. 46), thus getting O^h. Then point O^h will be the center of the ellipse which represents the center circle of the guide pulley in its actual position. Point 2, where K S cuts M^h D^h, will be one end of the minor axis, and point 1, found by laying off O^h 1 equal to O^h 2, will be the other end of the minor axis. The major axis is found by drawing a line through O^h parallel to M^h D^h and laying off along this line from O^h the distances O^h 3 and O^h 4 each equal to the radius of the guide pulley. Having now found the two axes of the center ellipse, it can be drawn by any geometric method for constructing an ellipse. We will next find O^v by prolonging the major axis of the ellipse just found until it meets the ground line at o^2, then erecting a perpendicular to the ground line at o^2 to meet V P at o^3, through o^3, drawing a line parallel to the ground line, and from O^h drawing a line perpendicular to the ground line which shall meet the parallel through o^3 at O^v. This point will be the vertical projection of the center of the middle circle of the guide pulley. The ellipse, which is the vertical projection of this middle

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*This subject is optional.

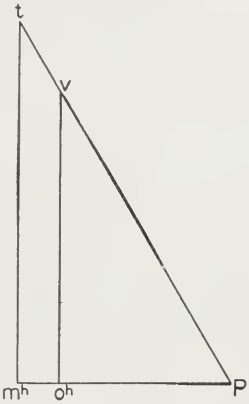


Fig. 47.

circle, is found from O^v in a way exactly similar to that in which the ellipse for the horizontal projection was found from O^h.

The next step is to draw the guide pulley and its shaft, and to do this we will revolve the center ellipse over in each view in such a way that we will have it projected as a line. We will take the horizontal projection first. Extend the major axis of the ellipse from O^h to o^4, making O^h o^4 equal to the perpendicular distance of O^v from the ground line. Draw O^6 1 through point 1 parallel to O^h o^4. With o^4 as a center and with a radius equal to the radius of the guide pulley, cut o^6 1 at o^6, and cut D^h M^h extended at o^5. Points o^6, o^4 and o^5 will be in a straight line, and the line joining them will be the edge view of the center circle of the guide pulley. About this line o^6 o^5 draw a rectangle as shown, the width of the rectangle being made equal to the width of the face of the guide pulley. Through o^4 draw a line perpendicular to o^6 o^5, which will be the revolved position of the center line of the guide-pulley shaft. The method of revolving back to get the axes of the ellipses, which are the projections of the edges of the pulley in its actual position, and to get the projection of the shaft, will be clear from a careful study of the figure.

The vertical projection of the guide pulley is found by revolving over in exactly the same way, the distance $O^v o^7$ being equal to the perpendicular distance of O^h from the ground line. It is well to assume a definite length for the shaft, whether this be the actual length which the shaft would be made or not. The length assumed in the figure is T^1 T^2 (same as T^3 T^4), and half of this is laid off each side of o^4 in the revolved horizontal projection.

This completes the two elevations of the guide pulley. The plan is drawn as follows: find the projection L L of the two ends of the shaft as shown by the construction lines, then revolve over by drawing line K K^1 at any convenient place parallel to L L, drawing perpendiculars through the two points L meeting the parallel line at K and K^1, and laying off on these perpendiculars the distances K^1 L^1 and K L^2 equal respectively to L^D P^D and L^E P^E in the elevation. The line L^1 L^2, joining the points L^1 and L^2 thus found, is the revolved position of the shaft, and should be equal in length to T^1 T^2 and T^3 T^4 in the elevations.

We can now draw the rectangle which represents the revolved position of the guide pulley at the middle of the line L¹ L², and find the ellipses from this rectangle in the same way as we found the ellipses from the rectangle in the other two views.

The belt is drawn in accordance with the same kind of reasoning as was used in determining the way the belt would look in the other kinds of quarter-turn belts which we have studied.

Belts connecting nonparallel shafts whose axes intersect. It very often happens that a belt must connect two shafts which are on the same level, but which are not parallel. The connection can be made, whatever the angle between the shafts, by the use of two guide pulleys. If the two main shaft pulleys are of the same diameter, the belt may be made to run in either direction by putting both guide pulleys, or mule pulleys as they are often called, on the same shaft, which will be perpendicular to the plane which contains the axes of the main shafts. That is, if the main shafts are horizontal, the shaft for the guide pulleys will be vertical. If the main pulleys are of different diameters, the guide pulleys may still be placed on the same vertical shaft, but in this case the belt can run in only one direction. If the belt is to run in either direction, the guide pulleys are placed on separate shafts, which are usually adjustable in position, so that they may be tipped at the proper angle to receive and deliver the belt.

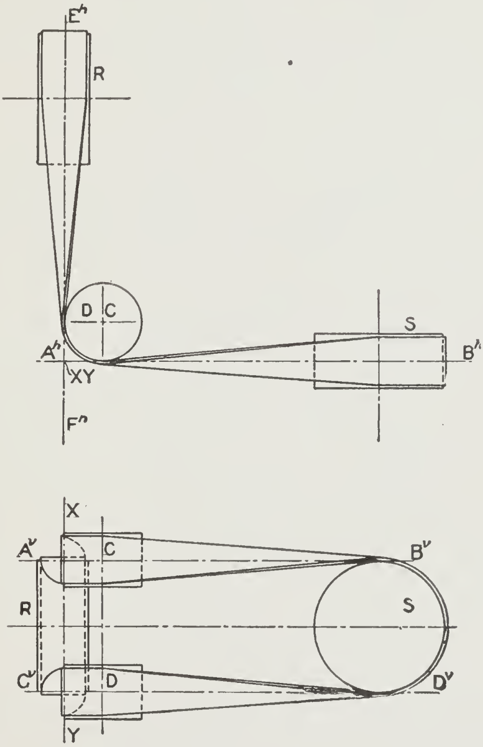


Fig. 48.

We will first take the case where the two main pulleys are of the same size. Fig. 48 shows the arrangement, the upper view being the plan and the lower view the elevation. R and S are the two main pulleys, and C and D are the two guide pulleys. The line X Y is the line of intersection of the planes of the main pulleys. The location of the guide pulleys is sufficiently clear from the drawing without further explanation.

Fig. 49 shows the arrangement when the main pulleys are of different diameters. Here the pulleys can turn only in the direction shown by the arrows, for if the direction were reversed the belt would leave the pulleys. The pulleys in plan appear the same as in Fig. 48. In the elevation the upper guide pulley, which receives the belt from S, has its center plane tangent to the pulley S at the point where the belt leaves S, as shown by the line A^v B^v. The lower guide pulley, which receives the belt from R, has its center plane tangent to R, as shown by C^v D^v. If the pulleys were to turn in the opposite direction, the upper guide pulley would have its plane tangent to R and the lower one have its plane tangent to S.

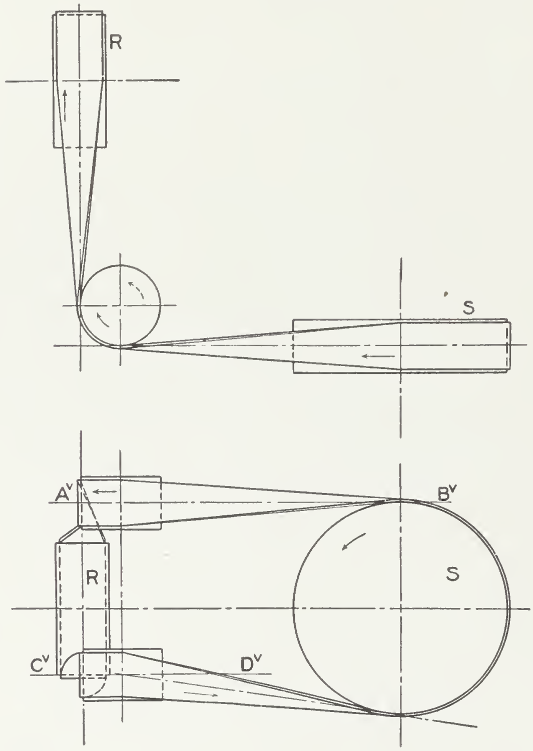


Fig. 49.

Belt Holes. Very often a belt has to pass through a floor or partition. The holes through which the belt runs should be large enough to be sure that the belt will never strike the sides, but it is desirable that they should be no larger than is necessary to accomplish that result. Accordingly, the holes should be laid out so that they may be cut in the right place and at the proper angle, the first time. Figs. 50 to 52 show the method of getting the position of the belt holes for the various kinds of belts. The top of the floor only is shown, as it is all that would need to be used in drawing the belt holes, the holes being cut through the floor parallel to the center line of the belt.

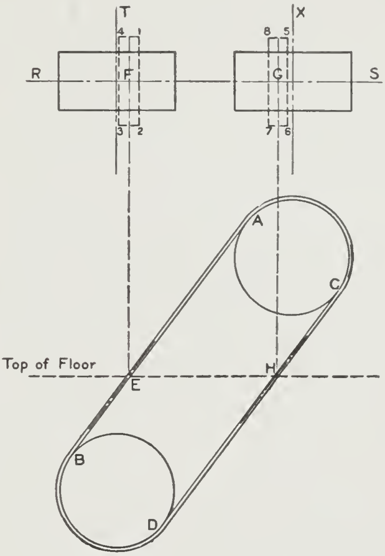


Fig. 50.

In Fig. 50 we have a common open belt. The circles representing the pulleys are drawn, and the belt drawn around them. A short piece of the pitch line is also drawn in each part of the belt where it passes through the floor. These parts of the pitch line are simply lines parallel to, and halfway between, the lines which represent the outer and inner faces of the belt. Next draw the two rectangles which represent the plan view of the pulleys, and draw through them the center line R S. From the points E and H, where the pitch line intersects the line which represents the top of the floor, draw perpendiculars to R S meeting it in points F and G. F and G are the center points of the rectangles 1 2 3 4 and 5 6 7 8, which form the outline of the belt holes on the surface of the floor. The long dimension of the rectangles will be parallel to the shafts on which the pulleys are located. After the belt holes are so found, the distances of their center lines to the right or left of the lines T and X respectively (which are the center lines of the shafts), can be measured on the drawing, and the workman can mark them out on the floor by plumbing down (or up) from the shafts, getting the lines T and X on the floor directly under or over the center of the

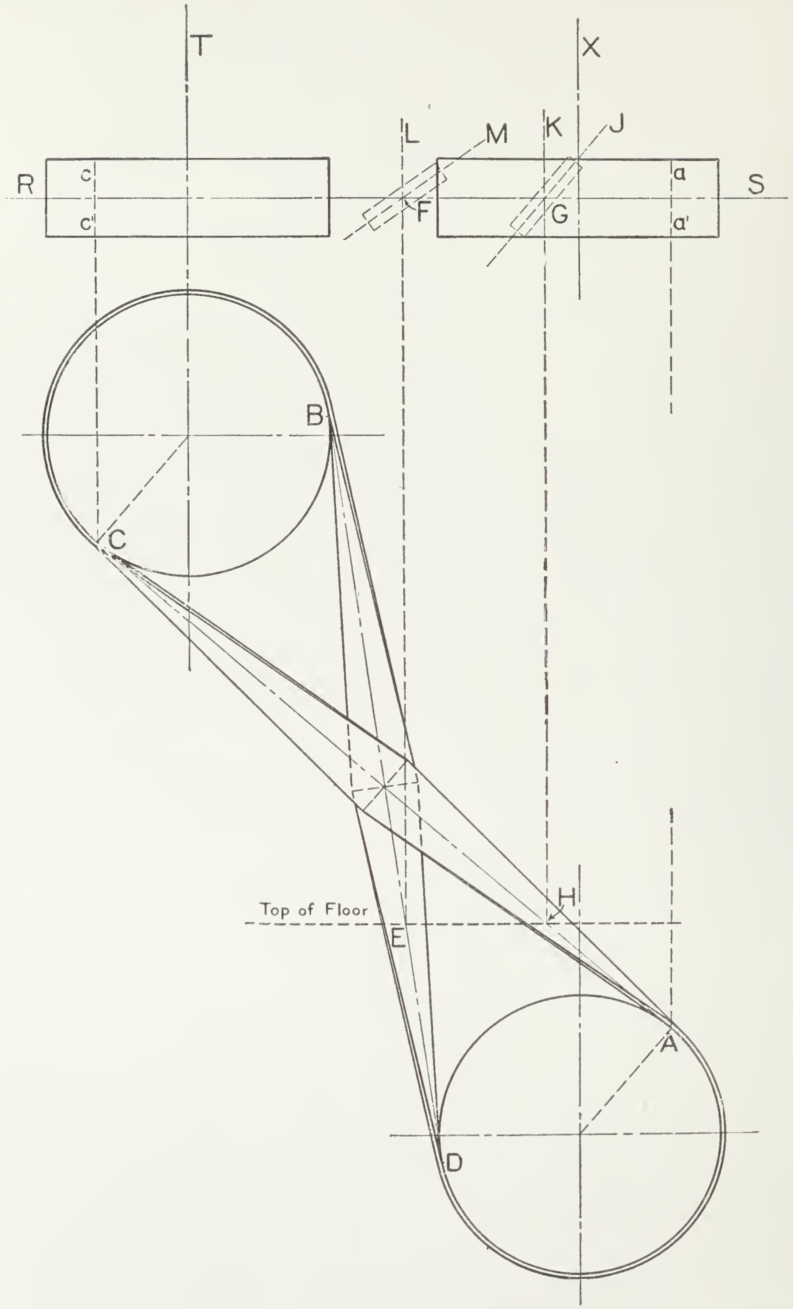


Fig. 51.

shafts, and thus locate on the floor the points F and G, and consequently the belt holes, from the dimensions taken from the drawing.

Fig. 51 shows how to draw the holes for a crossed belt. Draw the two views of the pulleys and the center lines A C and D B of the belt in the elevation ; also the center line R S in the plan. It is well also to draw the belt complete in the elevation, as it makes it easier to determine which way the belt holes will slant. From points E and H, where the center lines of the belt intersect the floor line, draw E L and H K perpendicular to R S and meeting R S in F and G. The points F and G are the center points of the belt holes, and it only remains to determine the angles which the center lines of the holes make with T and X respectively. These will be the same as the angles made with H K and E L. When the belt is leaving the pulley at A, a line drawn perpendicularly across to its inner face would occupy the position indicated by the dotted line *a a′* in plan, and the belt in passing from A to C twists through an angle of 180 degrees, and the line which was at *a a′* will occupy the position *c c′*. Therefore, when the belt has passed from A to H it will have twisted through an angle which will bear the same relation to 180 degrees that the distance A H bears to the distance A C. That is, if A H = $\frac{1}{4}$ of A C, the angle J G K is $\frac{1}{4}$ of 180 degrees, or 45 degrees. Whether the angle J G K shall be laid off to the right or to the left of line H K, must be reasoned out by considering which way the belt twists in passing from A to C. The angle of the other belt hole (L F M) is determined in the same way.

Fig. 52 shows the method of finding the belt holes of a plain quarter-turn belt, similar to Fig. 50. The centers G and F in plan are found by projecting from the elevation, as shown by the construction lines. The angle which the center line of hole at G makes with the center line of the shaft is found by dividing 90 degrees in the ratio of the distances P and N. The angle of the center line of the belt hole at F with the center line of the shaft is found in a similar manner, by dividing 90 degrees in the ratio of the distance A E to E C. It is usually sufficiently accurate, however, after having found the angle at G, to draw the center line of the other hole parallel to it.

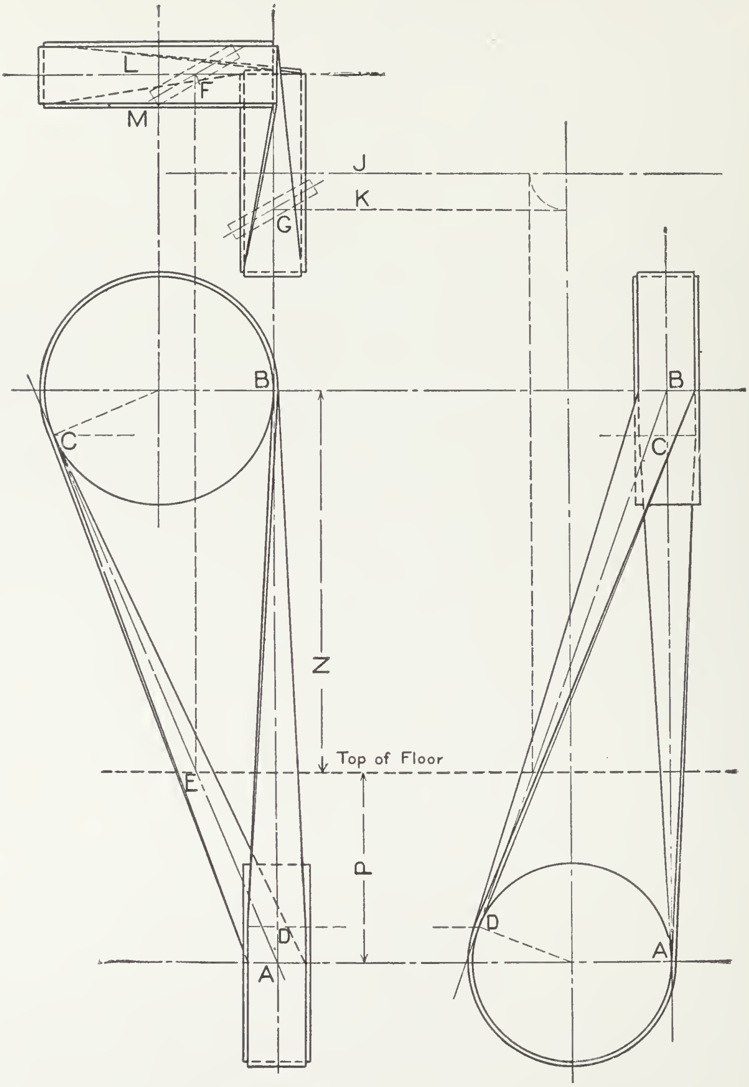


Fig. 52.

TOOTHED GEARING.

If we have a shaft B (Fig. 53), which has fastened to it a circular disc with a smooth circumference of radius B P, and parallel to and at a distance B A from the first shaft, another shaft A, which carries a disc of radius A P, it is evident that the two discs will touch each other at the point P, and if there were sufficient friction between the surfaces of the two discs, rotation of one shaft would cause rotation of the other. The number of turns that A would make would be to the number of turns that B would make as distance B P is to distance A P. In practice, however, the friction between the two discs would not be enough to transmit much power, so some more reliable means must be used.

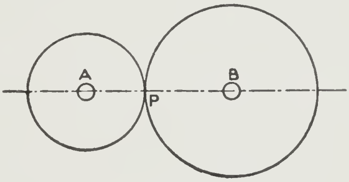


Fig. 53.

Wheels having teeth on their circumferences, as shown in Fig. 54, are used in place of these discs. The two dot and dash circles drawn about the centers A and B and in contact at the point P, correspond to the discs of Fig. 53, and the gears may be said to be based on these circles, which are called the *pitch circles.* The diameters of these circles are called the pitch diameters of the gears. The point P, where the two pitch circles touch each other, is called the pitch point. Circles drawn through the outer ends of the teeth are called the addendum circles, and circles drawn at the bottom of the teeth are called the root circles. That part of the tooth outline between the pitch circle and the addendum circle, as P E, is called the face of the tooth, and the part between the pitch circle and the root circle, as P F, is called the flank of the tooth. The distance between the addendum circle and the pitch circle, measured on a radial line, is the addendum, and the distance between the root circle and the pitch circle, measured on a radial line, is the root. The root is made a little greater than the addendum, the difference between the two being the clearance. The distance from the center of one tooth to the center of the next, measured on the pitch circle, as G H, is the circular or circumferential pitch, and is equal to the circumference of the pitch circle divided by the number of teeth. In order to run together, two gears must have the same circular pitch. The numbers of teeth in two gears of the same pitch are proportional to the circumferences and, consequently, the diameters of their pitch circles; and as the speeds of the two shafts which are connected by a pair of gears are inversely proportional to the diameters of the pitch circles of the gears, the speeds must also be inversely proportional to the number of teeth in the gears.

On rough gears the width of the tooth (as L M) is made a little less than the width of the space (R L), to allow for irregularities of construction. The difference in the width of the two is called the *backlash.*

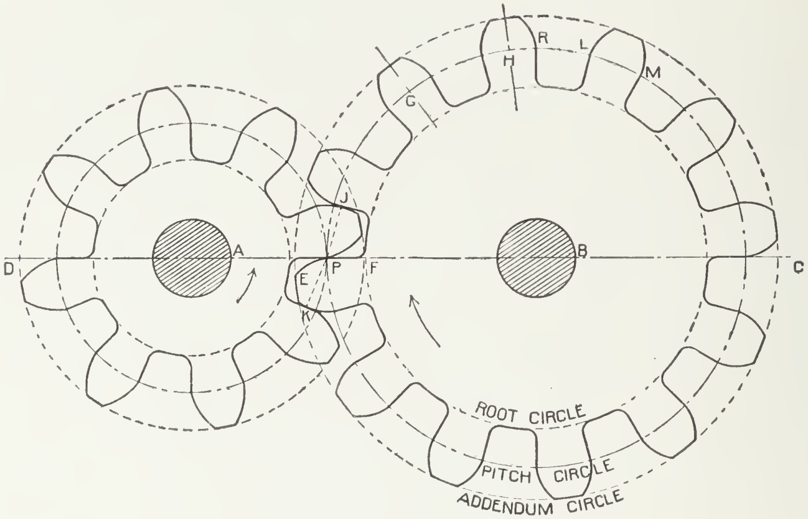


Fig. 54.

Although the circular pitch is a term which is used frequently in connection with gearing, there is another kind of pitch which is often used. This is the *diametral pitch*, and is equal to the diameter of the pitch circle divided by the number of teeth. The diametral pitch is not a distance which can be measured on the gear, as can the circular pitch, but is merely a term which is used to describe the size of the gear, and which bears the same relation to the circular pitch that the diameter of a circle bears to its circumference. Since the circumference of a circle is 3.1416 times its diameter, the circular pitch can be found by multiplying the diametral pitch by 3.1416, or the diametral pitch can be found by dividing the circular pitch by 3.1416. In speaking of the pitch of a gear, instead of using the diametral pitch, which is often a fraction, it is customary to use only the denominator of the fraction. For instance, if the diametral pitch of a gear is $\frac{1}{2}$ inch, it would be spoken of as a 2-pitch gear.

If the gear A (Fig. 54) is driving the other gear in the direction indicated, there is a working point of contact between the teeth of the two gears at J and another at K; and as the gears turn and the teeth slide along each other, the point where a pair of teeth are in contact changes. Now, if from the point of contact, as J, a line is drawn to the pitch point P, the tooth curve should be of such form that this line is normal to the curve at the point of contact. That is, P J should be perpendicular to a line drawn tangent to the tooth curve at J. This condition should hold wherever the point of contact of the two teeth is, in order that the gears shall run smoothly and with a uniform velocity. A short way of stating this principle is: The normal to the tooth curves, at the point where they are in contact, must always pass through the pitch point.

There are two kinds of curves which are commonly used for the outlines of the gear teeth, and which fulfill the above condition. These curves are the cycloidal and the involute curves, the drawing of which was explained in Part II. Although ordinarily a draughtsman is not required to draw the exact outline of gear teeth, yet every draughtsman and every one who has to do with machinery, should understand the principles of gears; and the best way to become familiar with those principles is by studying the method of drawing the various kinds of gears.

Epicycloidal Spur Gears. Fig. 55 shows a pair of epicycloidal gears designed to run together. The centers of the gears are at B and A; the pitch circles are shown in dot and dash, and are in contact at the pitch point P. The circle whose center is C, shown dotted, by rolling on the inside of the pitch circle of the gear B, generates the hypocycloid P E, which forms the flanks of the teeth on gear B; and by rolling on the outside of the pitch

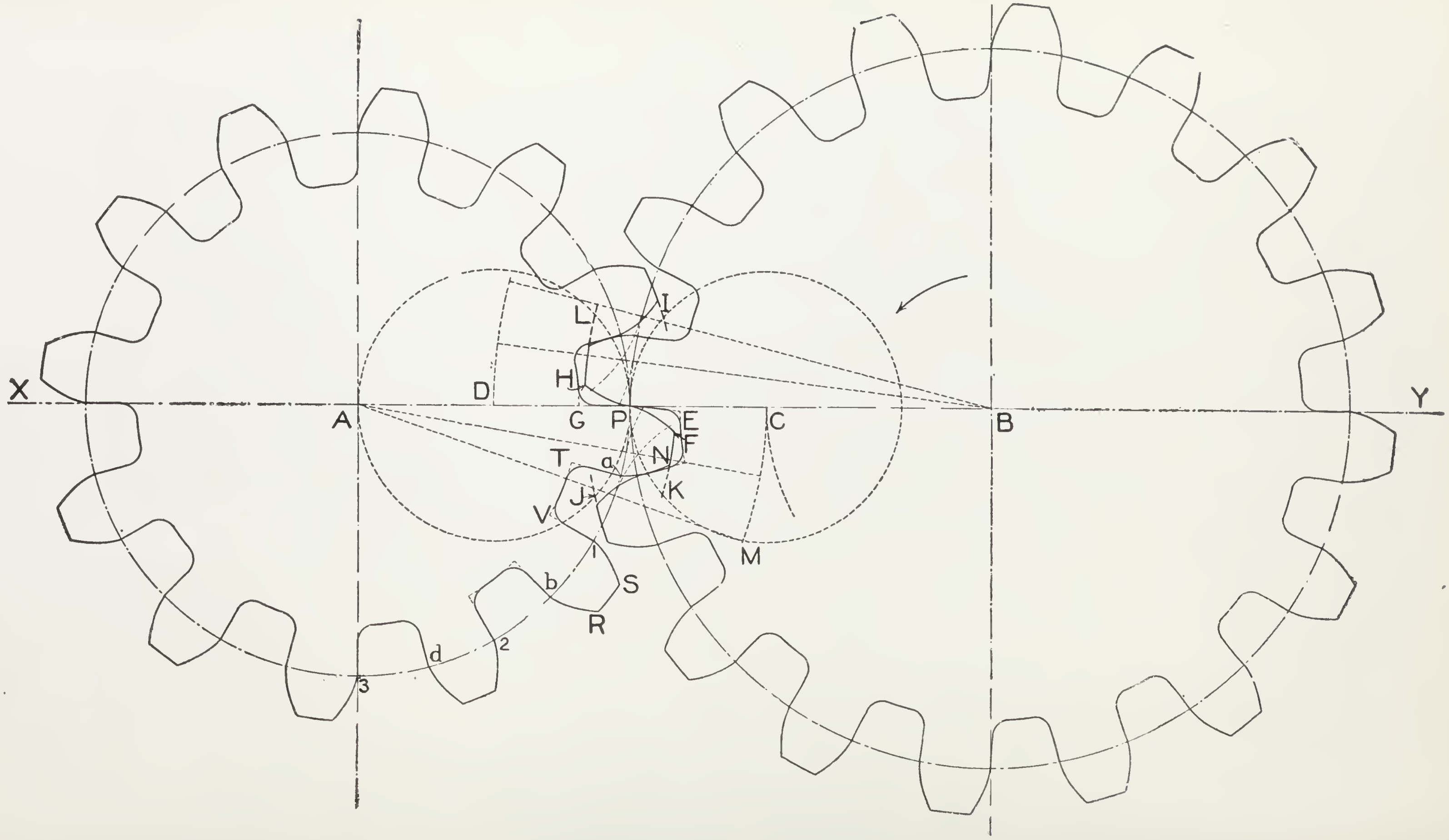


Fig. 55.

circle of gear A, generates the epicycloid P F, which forms the faces of the teeth on gear A. In like manner the circle whose center is D, by rolling on the inside of the pitch circle of gear A, generates the hypocycloid P G, which forms the flanks of the teeth on A, and by rolling on the outside of the pitch circle of B, generates the epicycloid P H, which forms the faces of the teeth on B. The circles C and D are called the describing circles. If the gear B is the driver and is turning in the direction shown by the arrow, the flanks of its teeth act on the faces of the teeth on A from the point where they first come in contact until the point of contact reaches the pitch point; and from the pitch point on until the contact ceases, the faces of the teeth of B act on the flanks of the teeth of A. In other words, the hypocycloidal part of the tooth curve on one gear is generated by the same describing circle that generates the epicycloidal part of the tooth on the other gear, with which it is in contact. This must always hold true in order to have the gears run properly. The arc I P of the describing circle C, together with the arc J P of the describing circle D, forms what is called the path of contact; that is, the point of contact between the teeth is always somewhere on the line I P J. If the gear A were the driver, the direction of rotation remaining the same, the path of contact would be L P K.

To design a pair of epicycloidal spur gears we must have given the pitch (either diametral or circular), the diameters of the pitch circles, or the number of teeth, and something to determine the size of the describing circles. Manufacturers have found by experience what are the best ratios of describing circles to pitch circles, and gears are designed according to those ratios. It is not well to have the diameter of the describing circle greater than $\frac{5}{8}$ the diameter of the pitch circle, and it is better to have it smaller. If a set of gears is to be made, any one of which is likely to run with any other one, the same size describing circle must be used for the faces and flanks of all the gears; and this describing circle is often taken $\frac{5}{8}$ the diameter of the smallest gear of the set. Sometimes when two gears are not part of an interchangeable set, but are designed to run with each other only, the diameter for the describing circle for the flanks of each gear is made equal to the radius of that gear, and when this is the case the flanks are radial straight lines, or, as it is usually stated, the gears have radial flanks.

In Fig. 55 the two describing circles are of the same size and equal to the radius of the smaller gear, thus giving radial flanks on this gear. Let us proceed with the design of this pair of gears, given dimensions as follows: gears to be 4 pitch (that is, as explained previously, four teeth per inch of pitch diameter); gear A to have 12 teeth; gear B, 16 teeth; addendum equal to the diametral pitch; clearance equal to $\frac{1}{8}$ of the addendum; describing circles each equal to radius of gear A. The steps in the process of drawing the gears are as follows:

First.— Calculate the diameters of the pitch circles.

Second.— Draw the center line X Y on the paper, and on this center line locate the centers A and B a distance apart equal to $\frac{1}{2}$ the sum of the two pitch diameters. Around these centers draw the pitch circles of diameters as calculated. This will make the pitch circles tangent at the pitch point P.

Third.— Calculate the radius of the addendum circles by calculating the addendum, and adding this amount to the radii of the pitch circles. Then draw the addendum circles with these radii.

Fourth.— Draw the root circles with radii equal to the radii of the pitch circles minus an amount equal to the addendum plus the clearance.

Fifth.— Draw the describing circles tangent to each other and to the pitch circles at the point P.

Sixth.— With the describing circle C rolling on the outside of the pitch circle of A, generate the epicycloid P F, continuing it until it meets the addendum circle of A. With the describing circle D rolling on the inside of the pitch circle of A, on the *opposite* side of line of centers from which the circle C rolled, generate the hypocycloid P G. Since the diameter of D is equal to the radius of the pitch circle of A, the hypocycloid P G will be a radial line; and consequently, after the student has become familiar with this fact, it will not be necessary to actually roll the circle to generate such a hypocycloid. The epicycloid P F and the hypocycloid P G together form one side of the tooth of gear A:

Seventh.— Divide the circumference of the pitch circle into as many equal parts as the gear has teeth, and through these points draw curves like the curve G P F. This may be done by making a templet of stiff paper that will just fit the curve G P F, and by means of this templet transferring the curve to the points 1, 2, 3, etc. Next find the points *a*, *b*, *d*, etc., halfway between 1 and 2, 2 and 3, etc., since there is to be no backlash, and through these points draw curves similar to G P F, but turned so as to curve the other way. Now by filling in with full lines that part of the addendum circle between the points F and N, R and S, etc., and filling in the root circle between T and V, etc., we have the outline of the teeth on the gear A. In practice, instead of making sharp corners at T and V, as shown by the dotted lines, fillets are put in with arcs of circles, these fillets being made as large as possible and still allow space so that the corner of the teeth on the other gear shall not strike.

Eighth.— Construct the teeth on the gear B in the same way as the teeth on A were constructed, the describing circle D generating the epicycloid P H by rolling on the outside of the pitch circle of B, and the describing circle C generating the hypocycloid P E by rolling on the inside of the pitch circle of B. The hypocycloid is not a straight line in this case, as the diameter of C is not equal to the radius of the pitch circle of B.

The following is the method of calculating the diameters of the pitch circles in the above construction. Since the gears are 4 pitch, their diametral pitch is $\frac{1}{4}$ inch, and the pitch diameter of each gear will be the number of teeth in that gear multiplied by the diametral pitch. Then gear A, having 12 teeth, will have for its pitch diameter $12 \times \frac{1}{4} = 3$ inches; and gear B, having 16 teeth, will have for its pitch diameter $16 \times \frac{1}{4} = 4$ inches.

If the pitch diameters had been given instead of the number of teeth, the latter would have been found by dividing the pitch diameters by the diametral pitch.

ANNULAR GEARS.

An annular gear is a ring with teeth on the inside of it. Fig. 56 shows such a gear, with center at A, meshing with a small spur gear, often called the pinion. The method of drawing such a pair of gears is similar to that just described for two spur gears. Here the circle C by rolling on the *inside* of the pitch circle of A generates the *faces* of the teeth of A, and the circle D by rolling on the *outside* of the pitch circle of A generates the *flanks* of the teeth on A. The shape of the teeth of the pinion in the figure is found with the same describing circles placed at C′ and D′ to avoid mixing up the lines.

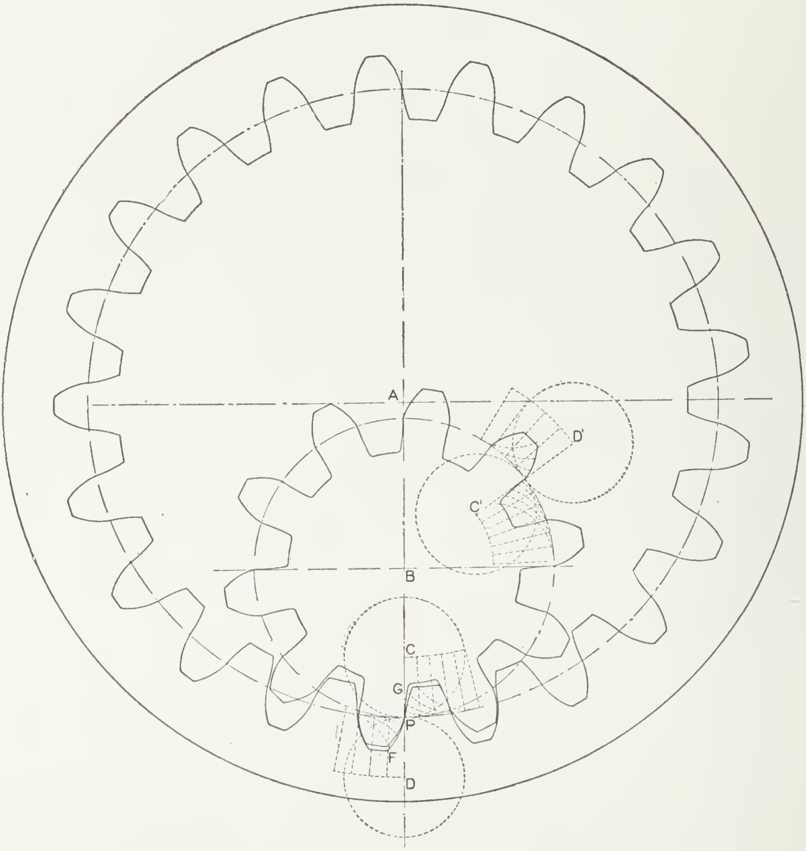


Fig. 56.

In designing an annular gear and pinion the diameter of the annular must never be so small that the distance from center A to center B shall be less than the sum of the radii of the two describing circles. If this should be the case, the teeth will interfere with each other.

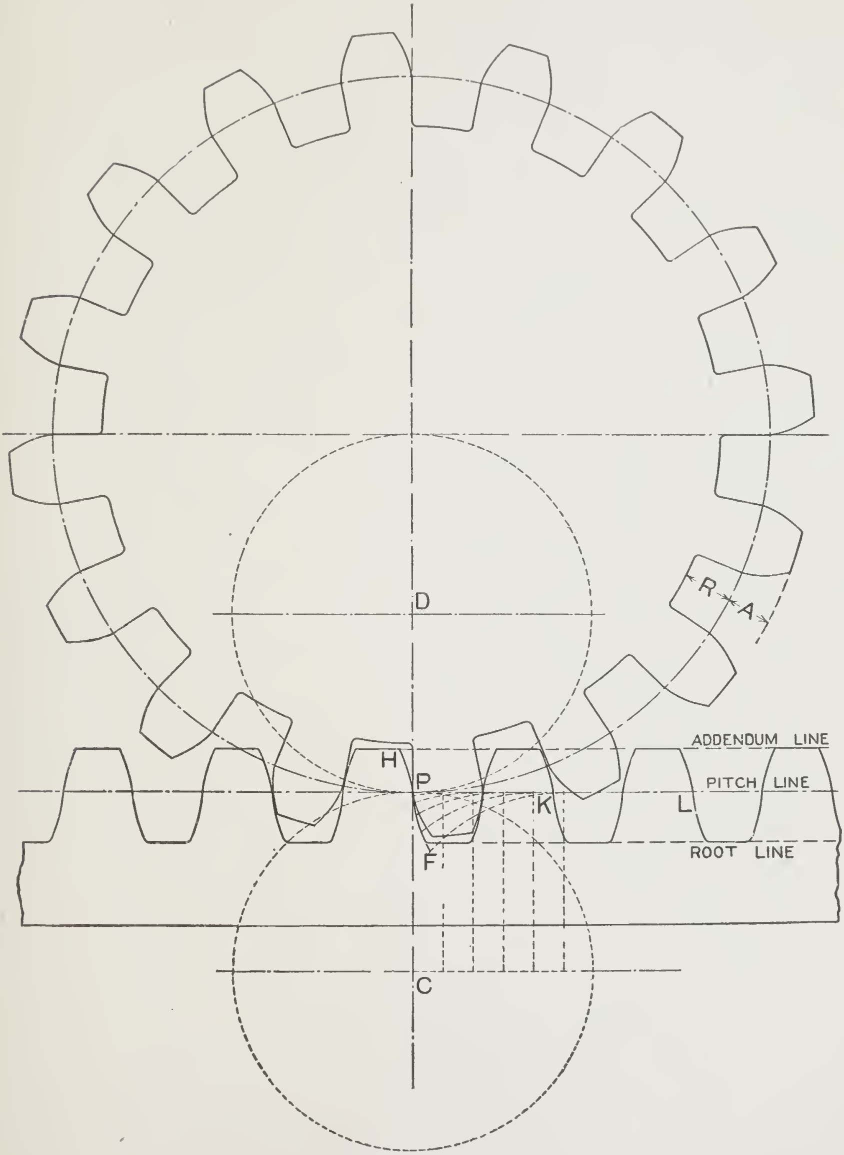


Fig. 57.

RACK AND PINION.

A rack is a gear whose pitch line is a straight line instead of a circle. Fig. 57 shows an epicycloidal rack in gear with a 16-tooth pinion. The describing circles are of the same size in the figure, although they might be of different sizes. The teeth on the pinion are drawn as described for Fig. 55, and the construction lines for drawing them are not shown. The curves which form the faces and flanks of the rack are cycloids. The describing circle C rolling on the pitch line of the rack, generates the cycloid P F, which forms the flanks of the rack teeth; and the describing circle D rolling on the pitch line of the rack, generates the cycloid P H, which forms the faces of the rack teeth. The addendum and root lines are drawn parallel to the pitch line, and at a distance from it equal respectively to the distances A and R of the pinion. The teeth are spaced off on the pitch line of the rack by laying off the distances P K, K L, etc., equal to the circular pitch of the pinion.

INVOLUTE GEARS.

Another form of gear teeth which is very common, is the involute. Fig. 58 shows parts of a pair of involute gears and the method of drawing the teeth. The circular pitch and the diameters of the pitch circles are found, and centers A and B located in the same way as described for epicycloidal gears. The pitch circles are in contact at the point P, as before. In involute gears the point of contact between the teeth is always somewhere on the line C D, the position of which is determined in designing the gears. This line passes through the pitch point, and the angle which it makes with the line X Y (the tangent to the pitch circles at P) is called the angle of obliquity, and is made what has been found by experience to give good results. It should not ordinarily be greater than 15 degrees. In designing the gears, having drawn the pitch circles, draw the line C D, making the desired angle with X Y, and with A and B as centers, draw circles tangent to this line. These circles are called the base circles. To find the addendum circles lay off on the line X Y the distances P M and P L, which, when possible, should each be as great as the circular pitch. From M and L draw lines perpendicular to C D,

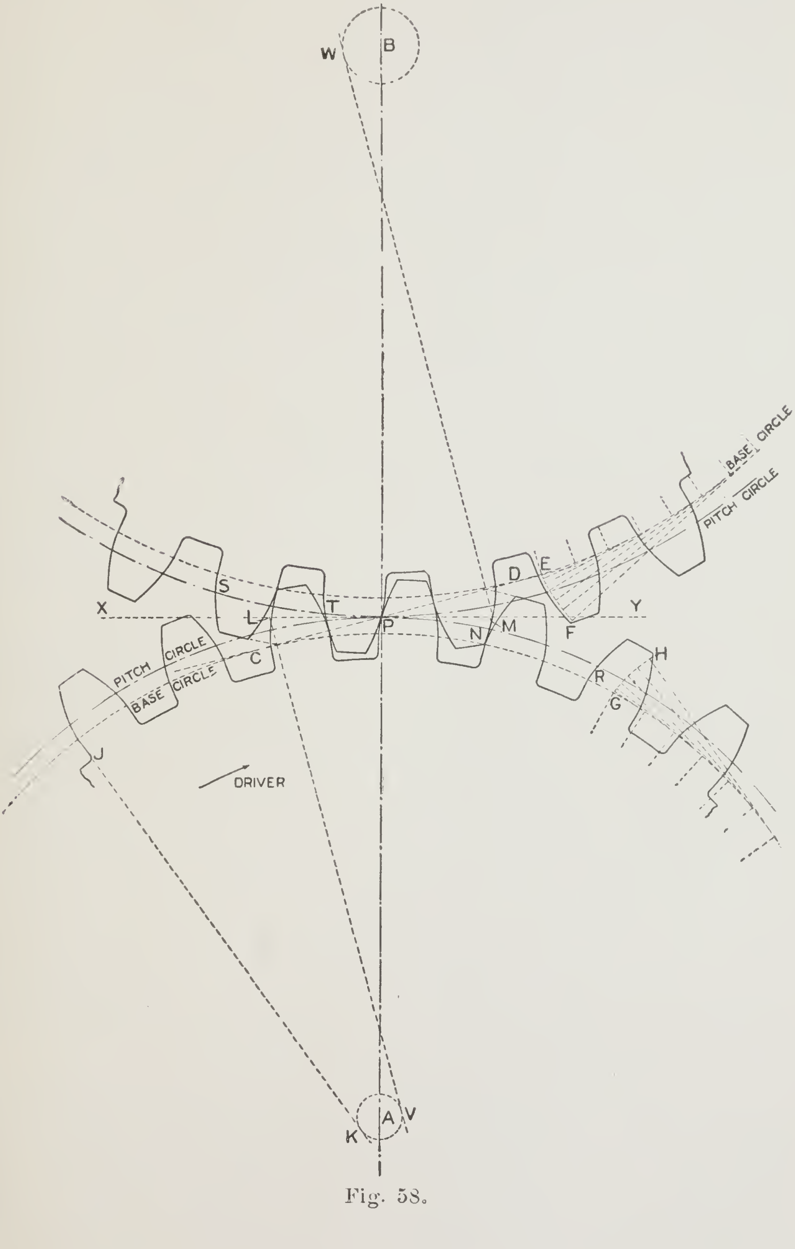

W
B
BASE CIRCLE
PITCH CIRCLE
X
S
T
D
E
Y
L
P
N
M
F
H
C
R
G
PITCH CIRCLE
BASE CIRCLE
J
DRIVER
K
A
V

Fig. 58.

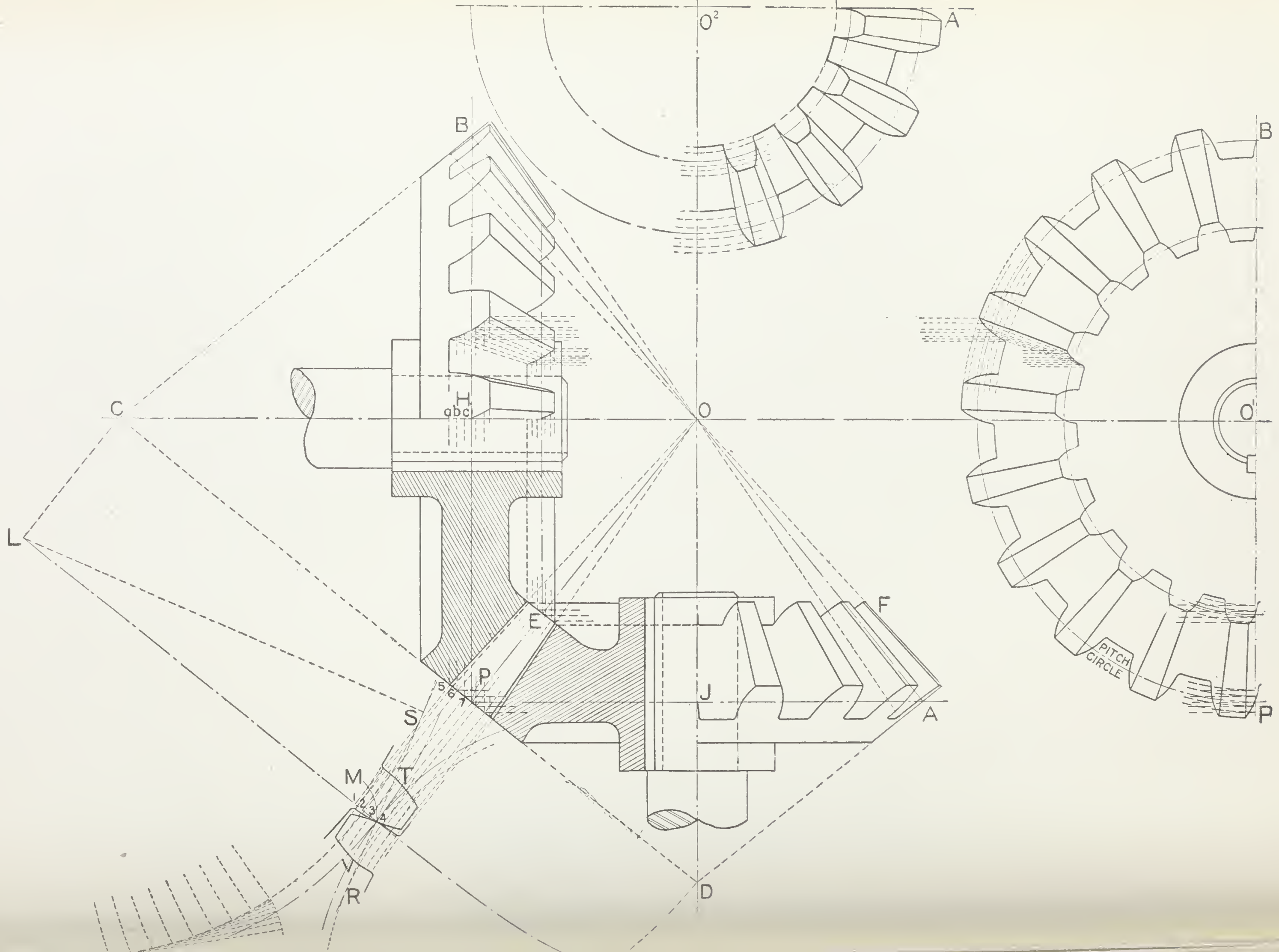
and draw the addendum circles through the points where these perpendiculars meet C D. The root circles are drawn by allowing a little clearance (about $\frac{1}{8}$ of the addendum). In laying off P M and P L care must be taken that neither one shall be so great that the point where the perpendicular strikes C D shall go outside of where C D is tangent to the base circle. P M plus P L must be at least equal to the circular pitch. At any convenient point on the base circles, as G and E, generate involutes in accordance with the methods described in Part II. Make templets to fit these curves, and by means of these templets draw in the tooth curves through P, N, R, S, T, etc., on the pitch circles, these points being found by spacing off the pitch just as described for epicycloidal gears. This gives us the working part of the teeth, and the remainder to the root circle is drawn by prolonging the perpendiculars, which were drawn through M and L, to V and W, and around the centers A and B respectively draw circles tangent to these lines. The flanks of the teeth are made straight lines tangent to the tooth curves already found and to these circles, as shown at J K. Small fillets are put in at the bottoms of the teeth.

BEVEL GEARS.

When two shafts whose axes intersect, are to be connected by gears, the gears commonly used are what are called bevel gears. The theoretically correct bevel gear is part of the sector of a sphere, but as the designing and making of such a gear is somewhat difficult, a cone is substituted for the sphere.

Fig. 59 shows a pair of *bevel gears* connecting two shafts whose axes intersect at right angles. The cones O P A and O P B are called the pitch cones, and the cones C P B and D P A the normal cones, and it is on these that the outlines of the teeth are laid out. The diameters B P and A P are the pitch diameters of the gears, and are found from the pitch and the number of teeth, just as the pitch diameters of spur gears are found.

To draw such a pair of gears we should have given the angle between the two shafts, the pitch and number of teeth in each gear (from which we find the pitch diameters), and the breadth of teeth P E. The gears might be either involute or epicycloidal. The addendum and root would be determined by empirical rules


O²
A
B
B
H
abc
C
O
O¹
L
E
F
5
6
7
P
S
J
A
M
T
1
2
3
4
V
R
D
PITCH CIRCLE
P

as on the other gears which have been discussed. The dimensions of the metal which forms the hub and the other parts of the gear, would be determined by the strength required, the place where the gear is to be used, etc.

The gears in Fig. 59 are 4 pitch, 16 and 20 teeth respectively. The addendum is as great as it can be. In this case it cannot be as great as the diametral pitch. The clearance is $\frac{1}{8}$ of the addendum. The gears are involute, with an angle of obliquity of 15 degrees. The shafts intersect at right angles. Start with point O and draw the center lines O C and O D, making an angle with each other equal to the given angle between the two shafts (in this case 90 degrees). Lay off on O C the distance O H equal to $\frac{1}{2}$ the pitch diameter of the smaller gear; through H draw a line perpendicular to O C. In like manner lay off on O D a distance O J equal to $\frac{1}{2}$ the pitch diameter of the large gear, and through J draw a line perpendicular to O D, meeting the perpendicular which was drawn through H at P. Make H B = H P, and J A = J P. From A, P and B draw lines to O, and we have the pitch cones. Through P draw C D perpendicular to O P, meeting O C and O D in C and D respectively. Join C B and D A and we have the normal cones. Through C, P and D draw perpendiculars. Lay off a distance C L = P M = D K. The lengths of C L, P M and D K may be anything convenient, provided they are all equal. With K and L as centers, and K M and L M as radii respectively, draw arcs of circles which will be tangent at M. These arcs will now be treated as pitch circles on which to design the tooth curves, being the development of portions of the normal cones. Through M draw the line of obliquity S R, and draw the base circles tangent to these lines. On the pitch circles lay off M V and M T each equal to $\frac{1}{2}$ the calculated circular pitch. If we should make the addendum equal to the diametral pitch, $\frac{1}{4}$ inch, the addendum circle of large gear would cut the line of obliquity beyond the point where R S is tangent to the base circle of the smaller gear, which we have previously learned should not occur, so we make the addendum circle of the large gear pass through R, and make the addendum of the small gear equal to the addendum of the large one. The root circles are drawn by allowing the clearance, as in the preceding gears.

One tooth on each gear is drawn as described for Fig. 58. With L as a center, draw a series of arcs (shown dotted), cutting the tooth which was drawn on the pitch circle which had L as a center, these same arcs cutting the line L K at points 2, 3, 4, etc.; from 2, 3, 4, etc., draw lines perpendicular to C D, cutting C D at 5, 6, 7, etc. From these points draw lines to O. Along P O lay off P E (which must be given), and through E draw a line perpendicular to P O, cutting 5 O, 6 O, etc. From the points of intersection draw other lines parallel to P B. With center O′ taken at any convenient place on C O prolonged, and with radii equal in turn to *a* 5, *b* 6, etc., draw circles as shown. On the circle which is drawn with H P as a radius (marked pitch circle), space off the circular pitch, and on each of the circles in turn lay off the teeth of the same width as they are on the corresponding circles drawn through 1, 2, 3, 4, etc. The rest of the construction can be understood from a careful study of the figure. The other gear is drawn in the same way. Half of each gear is drawn in section in the figure to show the details of the casting.

PLATES.

Plates XX to XXIV inclusive, are to be drawn by the student to give him practice in applying the principles which he has learned of screws, springs, cams, belts and gears. These plates are to be made the same size as those of the preceding Instruction Papers of the course; that is, 11 inches by 15 inches, outside, with a margin of $\frac{1}{2}$ inch, making the part of the plate in which the drawing is done 10 inches by 14 inches.

PLATE XX.

The arrangement of this plate is shown by the reproduced plate.

Fig. 1 is to be a drawing of a right-hand single V-threaded screw 4 inches diameter, 6 inches long, pitch of screw 1 inch. The true helix is to be constructed as shown in Fig. 2 of the Instruction Paper. Show dotted construction lines for one turn of the helix.

Fig. 2 is to be the conventional representation of a single right-hand square-threaded screw 2 inches diameter, $3\frac{1}{2}$ inches long, pitch $\frac{1}{2}$ inch (2 threads per inch).

SCREWS
AND
SPRINGS
EXACT AND CONVENTIONAL
REPRESENTATIONS

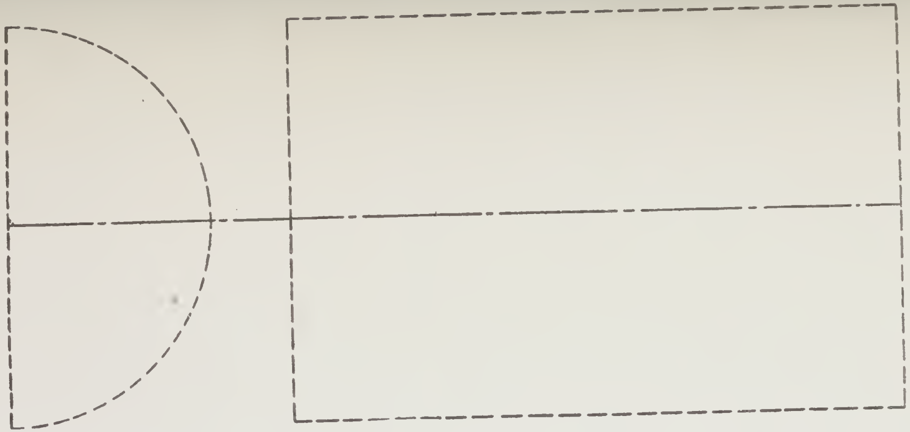


FIG 1

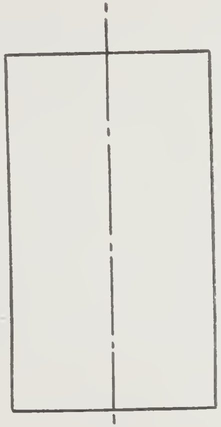


FIG. 2

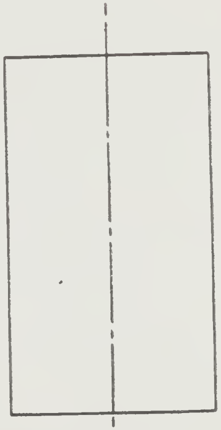


FIG. 3

FIG 4

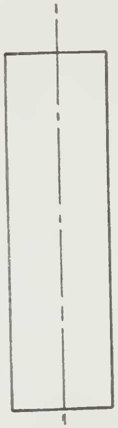


FIG. 5

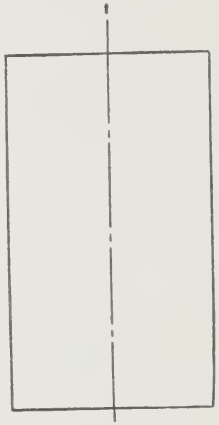


FIG. 6

JUNE 2, 1901. HERBERT CHANDLER CHICAGO, ILL.

PLATE XXI

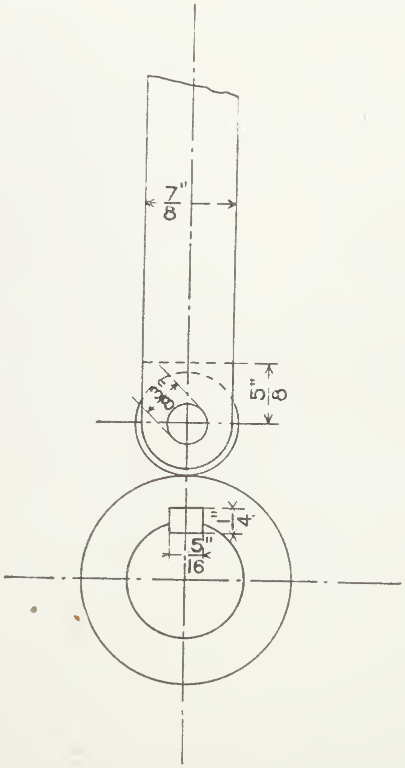


FIG. 1

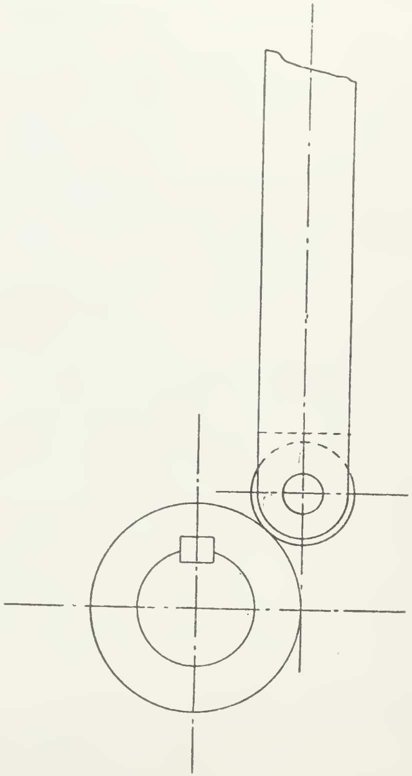


FIG. 2

JUNE 12, 1901. HERBERT CHANDLER CHICAGO ILL

Fig. 3 is to be the conventional representation of a single left-hand V-threaded screw 2 inches diameter, $3\frac{1}{2}$ inches long, $4\frac{1}{2}$ threads per inch. Use the conventional form shown in Fig. 23.

Fig. 4 is to be a single right-hand V-threaded screw 1 inch diameter, $3\frac{1}{2}$ inches long, 8 threads per inch. Use conventional form shown in Fig. 23.

Fig. 5 same as Fig. 4, except that the thread is to be a double thread, 4 threads per inch ($\frac{1}{4}$ inch pitch).

Fig. 6 is to be a right-handed helical spring of $\frac{5}{16}$ inch round steel, inside diameter of helix $1\frac{3}{8}$ inches, length $3\frac{1}{2}$ inches, pitch $\frac{5}{8}$ inch, as shown conventionally in Fig. 8.

PLATE XXI.

Fig. 1 is to be a plate cam, which, by turning right handed, is to raise a slide (having a roller at its lower end) $1\frac{1}{2}$ inches with harmonic motion while the cam is making one-half of a revolution, then allow it to drop downward to its original position, and remain at rest during the remaining one-half revolution. The center line of motion of the slide is to pass through the axis of the cam. The roller is to be 1 inch diameter, and at its lowest position its center is to be $1\frac{1}{2}$ inches above the center of the cam. The shaft of the cam is to be $1\frac{1}{8}$ inches in diameter and the hub 2 inches diameter. Show one view only. The details of the slide and roller of the hub of the cam are shown in the reproduced plate.

Fig. 2 is to be a plate cam, which, by turning left handed, is to raise a slide and roller, similar to that in Fig. 1, $1\frac{1}{2}$ inches with uniform motion during two-thirds of a revolution of the cam, and return it to its original position with gravity motion in the remaining one-third revolution. The center line of motion is 1 inch to the right of the axis of the cam, and at its lowest position the center of the roller is $1\frac{1}{2}$ inches away from the center of the cam. The dimensions for hub and for roller and slide are the same as for Fig 1.

PLATE XXII.

The reproduced plate shows plan and one elevation of two pulleys located on two shafts which are at right angles to each other. The student is to draw these two views and another elevation, showing the belt on the pulleys in all three views, and put on arrows to show which way the pulleys will have to turn in order that the belt may stay on the pulleys. The dimensions are given in feet and inches, but of course the drawing cannot be made full size, but must be scaled down. The scale, as stated in the table, is to be 1 inch for every foot; that is, if the distance between the shafts is 8 feet, we draw it 8 inches, and everything else in the same proportion. In other words, everything is drawn one-twelfth of its actual size.

PLATE XXIII.

Draw an 8-tooth epicycloidal spur gear meshing with a 20-tooth annular. The gears are to be 1 pitch, addendum of each gear equal to 1 inch, clearance $\frac{1}{8}$ inch. Diameter of both describing circles 4 inches. Draw all of the teeth on the 8-tooth gear, and as much of the annular as there is room for on the plate.

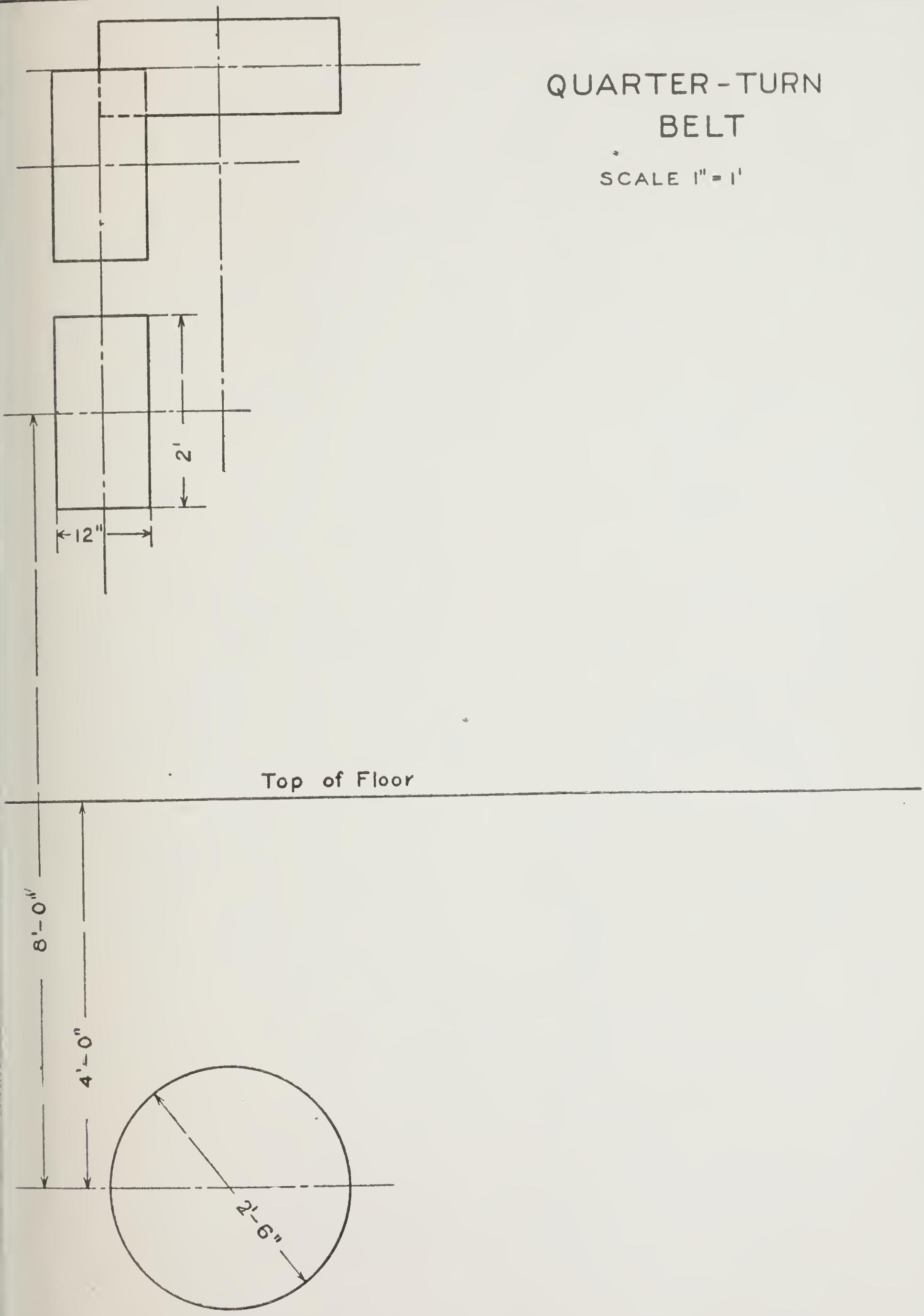
PLATE XXIV.

Draw a 24-tooth involute gear meshing with a 32-tooth involute gear. Gears to be 2 pitch, angle of obliquity 15 degrees, addendum $\frac{1}{2}$ inch, clearance $\frac{1}{16}$ inch. Show as much of each gear as there is room for on the plate.

PLATE XXII

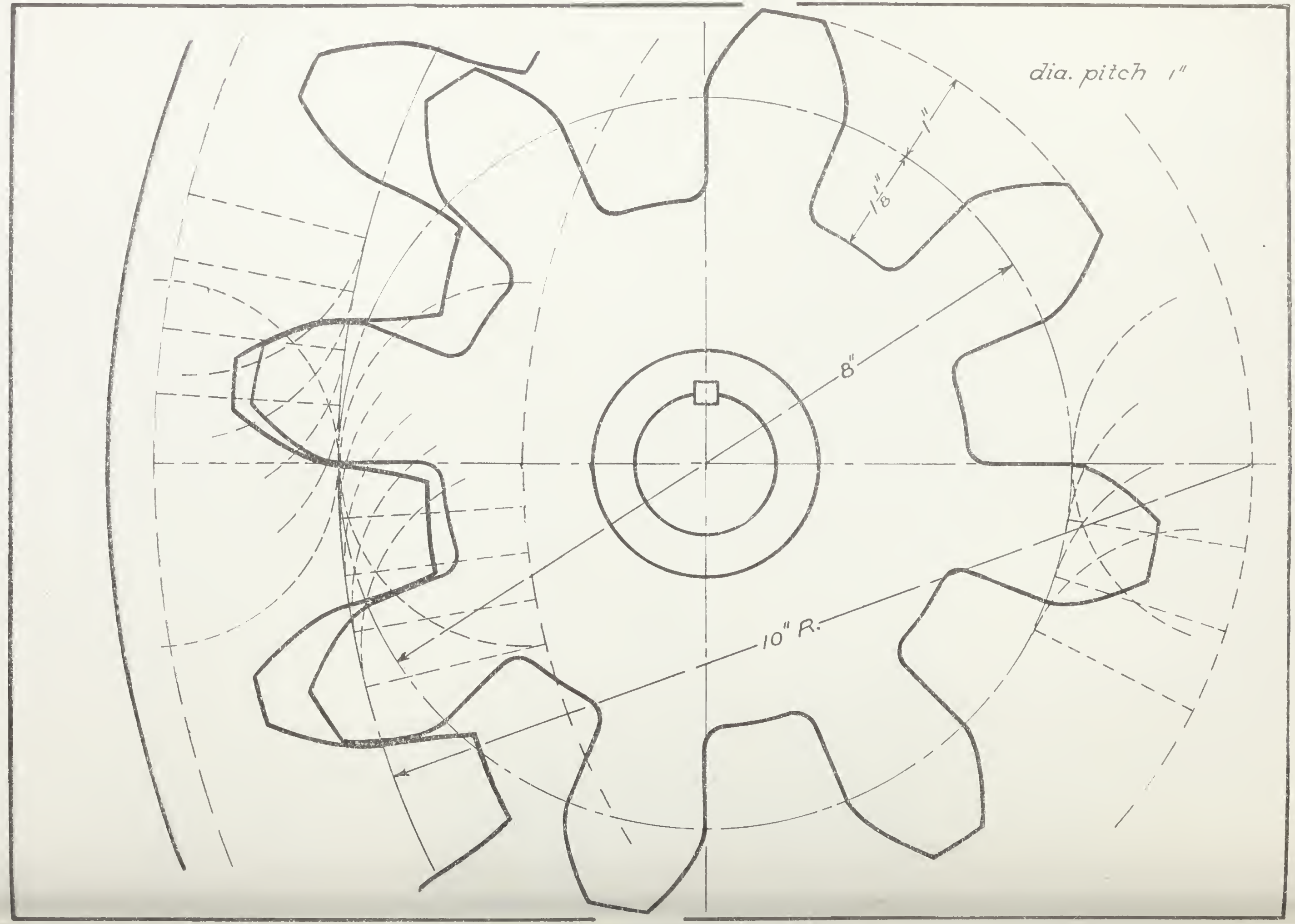
QUARTER-TURN BELT

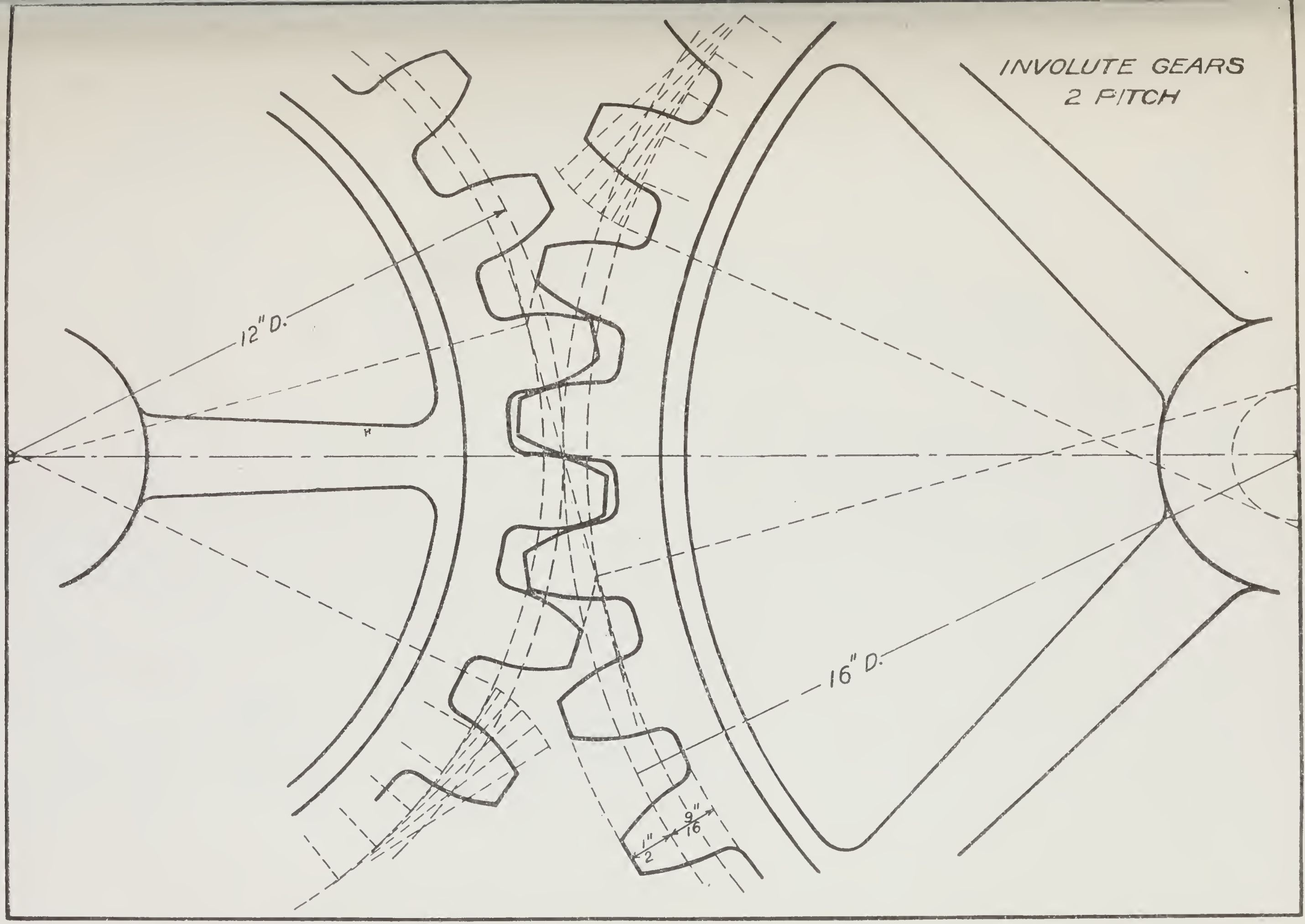
SCALE 1" = 1'



JUNE 20, 1901. *HERBERT CHANDLER* *CHICAGO, ILL.*

PLATE XXIII

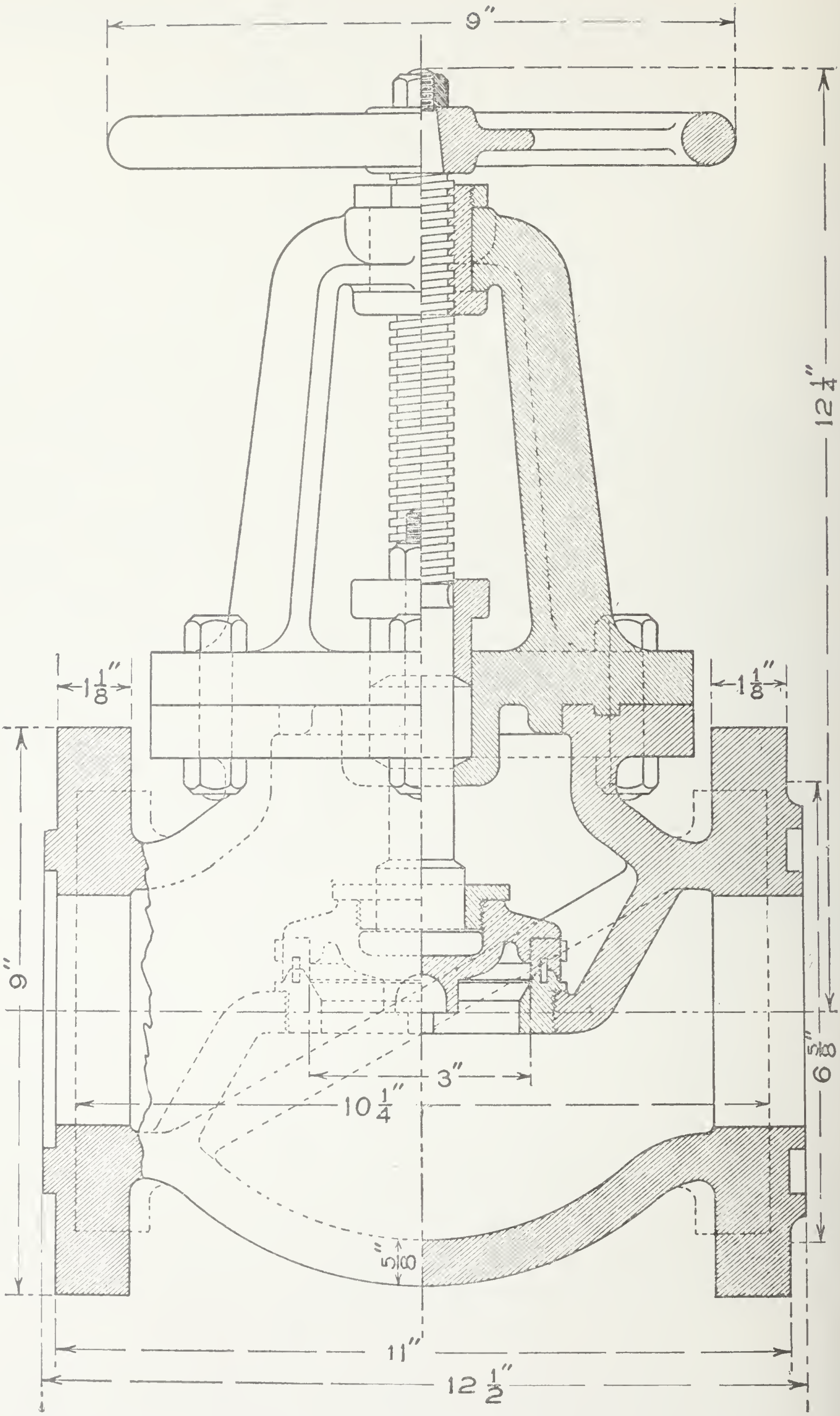




INVOLUTE GEARS
2 PITCH

JULY 7, 1901

HERBERT CHANDLER CHICAGO, ILL.



WORKING DRAWING SHOWING PRINCIPAL DIMENSIONS OF SPRING SEAT VALVE.

Crosby Steam Gage & Valve Co.

MACHINE DESIGN.

PART I.

Definition. Machine Design is the art of mechanical thought, development, and specification.

It is an art, in that its routine processes can be analyzed and systematically applied. Proficiency in the art positively cannot be attained by any "short cut" method. There is nothing of a spectacular nature in the methods of Machine Design. Large results cannot be accomplished at a single bound, and success is possible only by a patient, step-by-step advance in accordance with well-established principles.

"Mechanical thought" means the thinking of things strictly from their mechanical side; a study of their mechanical theory, structure, production, and use; a consideration of their mechanical fitness as parts of a machine.

"Mechanical development" signifies the taking of an idea in the rough, in the crude form, for example, in which it comes from the inventor, working it out in detail, and refining and fixing it in shape by the designing process. Ideas in this way may become commercially practicable designs.

"Mechanical specification" implies the detailed description of designs, in such exact form that the shop workmen are enabled to construct completely and put in operation the machines represented in the designs.

The object of Machine Design is the creation of machinery for specific purposes. Every department of a manufacturing plant is a controlling factor in the design and production of the machines built there. A successful design cannot be out of harmony with the organized methods of production. Hence in the high development of the art of Machine Design is involved a knowledge of the operations in all the departments of a manufacturing plant. The student is therefore urged not only to familiarize himself with the direct production of machinery, but to study the relation thereto of the allied commercial departments.

He should get into the spirit of business at the start, get into the shop atmosphere, execute his work just as though the resulting design were to be built and sold in competition. He should visit shops, work in them if possible, and observe details of design and methods of finishing machine parts. In this way he will begin to store up bits of information, practical and commercial, which will have valuable bearing on his engineering study.

The labor involved in the design of a complicated automatic machine is evidenced by the designer's wonderful familiarity with its every detail as he stands before the completed machine in operation and explains its movements to an observer. The intricate mass of levers, shafts, pulleys, gears, cams, clutches, etc., etc., packed into a small space, and confusing even to a mechanical mind, seems like a printed book to the designer of them.

This is so because it is a familiar journey for the designer's mind to run over a path which it has already traversed so many times that he can see every inch of it with his eyes shut. Every detail of that machine has been picked from a score or more of possible ideas. One by one, ideas have been worked out, laid aside, and others taken up. Little by little, the special fitness of certain devices has become established, but only by patient, careful consideration of others, which at first seemed equally good.

Every line, and corner, and surface of each piece, however small that piece may be, has been through the refining process of theoretical, practical, and commercial design. Every piece has been followed in the mind's eye of its designer from the crude material of which it is made, through the various processes of finishing, to its final location in the completed machine; thus its bodily existence there is but the realization of an old and familiar picture.

What wonder that the machine seems simple to the designer of it! As he looks back to the multitude of ideas invented, worked out, considered and discarded, the machine in its final form is but a trifle. It merely represents a survival of the fittest.

No successful machine, however simple, was ever designed that did not go through this slow process of evolution. No machine ever just simply happened by accident to do the work for which it is valued. No other principle upon which the successful design of machinery depends is so important as this careful, patient consideration of detail. A machine is seldom unsuccessful because some main point of construction is wrong. The principal features of a machine are usually the easiest to determine. It is a failure because some little detail was overlooked, or hastily considered, or allowed to be neglected, because of the irksome labor necessary to work it out properly.

There is no task so tedious, for example, as the devising of the method of lubricating the parts of a complicated machine. Yet there is no point of design so vital to its life and operation as an absolute assurance of an adequate supply of oil for the moving parts at all times and under all circumstances. Suitable means often cannot be found, after the parts are together, hence the machine goes into service on a risky basis, with the result, perhaps, of early failure, due to "running dry." Good designers will not permit a design to leave their hands which does not provide practically automatic oiling, or at least such means of lubrication that the operator can offer no excuse for neglecting to oil his machine. This is but a single illustration of many which might be presented to impress the definite and detail character necessary in work in Machine Design.

Relation. The relation which Machine Design should correctly bear to the problems that it seeks to solve, is twofold; and there are, likewise, two points of view corresponding to this twofold relation, from which a study of the subject should be traced. Neither of these can be discarded and an efficient mastery of the art attained. These points are—

I. Theory.
II. Production.

I. *Theory.* From this point of view, Machine Design is merely a skeleton or framework process, resulting in a representation of ideas of pure motion, fundamental shape, and ideal proportion. It implies a working knowledge of physical and mathematical laws. It is a strictly scientific solution of the problem at hand, and may be based purely on theory which has been reasoned out by calculation or deduced from experiment. This is the only sure foundation for intelligent design of any sort.

But it is not enough to view the subject from the standpoint of theory alone. If we stopped here we should have nothing but mechanisms, mere laboratory machines, simply structures of ingenuity and examples of fine mechanical skill. A machine may be correct in the theory of its motions; it may be correct in the theoretical proportions of its parts; it may even be correct in its operation for the time being; and yet its complication, its misdirected and wasteful effort, its lack of adjustment, its expensive and irregular construction, its lack of compactness, its difficulty of ready repair, its inability to hold its own in competition—any of these may throw the balance to the side of failure. Such a machine, commercially considered, is of little value. No shop will build it, no machinery house will sell it, nobody will buy it if it is put on the market.

Thus we see that, aside from the theoretical correctness of principle, the design of a machine must satisfy certain other exacting requirements of a distinctly business nature.

II. *Production.* From this point of view, Machine Design is the practical, marketable development of mechanical ideas. Viewed thus, the theoretical, skeleton design must be so clothed and shaped that its production may be cheap, involving simple and efficient processes of manufacture. It must be judged by the latest shop methods for exact and maximum output. It must possess all the good points of its competitor, and, withal, some novel and valuable ones of its own. In these days of keen competition it is only by carefully studied, well-directed effort toward rapid, efficient, and, therefore, cheap production that any machine can be brought to a commercial basis, no matter what its other merits may be. All this must be thought of and planned for in the design, and the final shapes arrived at are quite as much a result of this second point of view as of the first.

As a good illustration of this, may be cited the effect of the present somewhat remarkable development of the so-called "high speed" steels. The speeds and feeds possible with tools made of these steels are such that the driving power, gearing, and feed mechanism of the ordinary lathe are wholly inadequate to the demands made upon them when working the tool to its limit. This means that the basis of design as used for the ordinary tool steel will not do, if the machine is expected to stand up to the cuts possible with the new steels. Hence, while the old designs were right for the old standard, a new one has been set, and a thorough revision on a high-speed basis is imminent, else the market for them as machines of maximum output will be lost.

From these definitions it is evident that the designer must not only use all the theory at his command, but must continually inform himself on all processes and conditions of manufacture, and keep an eye on the tendency of the sales markets, both of raw material and the finished machinery product. This is what in the broadest sense is meant by the term "Mechanical Thought," thought which is directed and controlled, not only by theoretical principle but by closely observed practice. From the feeblest pretenders of design to those engineers who consummate the boldest feats and control the largest enterprises, the process which produces results is always the same. Although experience is necessary for the best mechanical judgment, yet the student must at least begin to cultivate good mechanical sense very early in his study of design.

Invention. Invention is closely related to Machine Design, but is not design itself. Whatever is invented has yet to be designed. An invention is of little value until it has been refined by the process of design.

Original design is of an inventive nature, but is not strictly invention. Invention is usually considered as the result of genius, and is announced in a flash of brilliancy. We see only the flash, but behind the flash is a long course of the most concentrated brain effort. Inventions are not spontaneous, are not thrown off like sparks from the blacksmith's anvil, but are the result of hard and applied thinking. This is worth noting carefully, for the same effort which produces original design may develop a valuable invention. But there is little possibility of inventing anything except through exhaustive analysis and a clear interpretation of such analysis.

Handbooks and Empirical Data. The subject matter in these is often contradictory in its nature, but valuable nevertheless. Empirical data are data for certain fixed conditions and are not general. Hence, when handbook data are applied to some specific case of design, while the information should be used in the freest manner, yet it must not be forgotten that the case at hand is probably different, in some degree, from that upon which the data were based, and unlike any other case which ever existed or will ever again exist. Therefore the data should be applied with the greatest discretion, and when so applied will contribute to the success of the design at least as a check, if not as a positive factor.

The student should at the outset purchase one good handbook, and acquire the habit of consulting it on all occasions, checking and comparing his own calculations and designs therefrom. Care must be taken not to become tied to a handbook to such an extent that one's own results are wholly subordinated to it. Independence in design must be cultivated, and the student should not sacrifice his calculated results until they can be shown to be false or based on false assumption. Originality and confidence in design will be the result if this course be honestly pursued.

Calculations, Notes, and Records. Accurate calculations are the basis of correct proportions of machine parts. There is a right way to make calculations and a wrong way, and the student will usually take the wrong way unless he is cautioned at the start.

The wrong way of making calculations is the loose and shiftless fashion of scratching upon a scrap of detached paper marks and figures, arranged in haphazard form, and disconnected and incomplete. These calculations are in a few moments' time totally meaningless, even to the author of them himself, and are so easily lost or mislaid that when wanted they usually cannot be found.

Engineering calculations should always be made systematically, neatly, and in perfectly legible form, in some permanently bound blank book, so that reference may always be had to them at any future time for the purpose of checking or reviewing. Put all the data down. Do not leave in doubt the exact conditions under which the calculations were made. Note the date of calculation.

If a mistake in figures is made, or a change is found necessary, never rub out the figures or tear out the leaf, or in any way obliterate the figures. Simply draw a bold cross through the wrong part and begin again. Often a calculation which is supposed to be wrong is later shown to be right, or the facts which caused the error may be needed for investigation and comparison. Time which is spent in making figures is always valuable time, time too precious to be thrown away by destroying the record.

The recording of calculations in a permanent form, as just described, is the general practice in all modern engineering offices. This plan has been established purely as a business policy. In case of error it locates responsibility and settles dispute. Consistent designing is made possible through the records of past designs. Proposals, estimates, and bids may often be made instantly, on the basis of what these record books show of sizes and weights. This bookkeeping of calculations is as important a factor of systematic engineering as bookkeeping of business accounts is of financial success.

The student should procure for this purpose a good blank book with a firm binding, size of page not smaller than 6 by 8 inches (perhaps 8 by 11 inches may be better), and every calculation, however small and apparently unimportant, should be made in it.

Sample pages of engineering calculations are reproduced in Figs. 3 to 9. Note the sketch showing the forces. Note the clear statement of data. Note the systematic writing of the equations, and the definite substitutions therein. Note the heavy double underscoring of the result, when obtained. There is nothing in the whole process of the calculation that cannot be reviewed at any moment by anybody, and in the briefest time.

The development of a personal note-book is of great value to the designer of machinery. The facts of observation and experience recorded in proper form, bearing the imprint of intimate personal contact with the points recorded, cannot be equalled in value by those of any hand or reference book made by another. There is always a flavor about a personal note-book, a sort of guarantee, which makes the use of it by its author definite and sure.

The habit of taking and recording notes, or even knowing what notes to take, is an art in itself, and the student should begin early to make his note-book. Aside from the value of the notes themselves as a part of his personal equipment, the facility with which his eye will be trained to see and record mechanical things will be of great value in all of his study and work. How many men go through a shop and really see nothing of the operations going on therein, or, seeing them, remember nothing! An engineer, trained in this respect, will to a surprising degree be able to retain and sketch little details which fall under his eye for a brief moment only, while he is passing through a crowded shop.

Some draftsmen have the habit of copying all the standard tables of the various offices in which they work. While these are of some value in a few cases, yet this is not what is meant by a good note-book in the best sense. Ideas make a good note-book, not a mere tabulation of figures. If the basis upon which standards are founded can be transferred to permanent personal record, or novel methods of calculation, or simple features of construction, or data of mechanical tests, or efficient arrangement of machinery—if *these* can be preserved for reference, the note-book will be of greatest value.

Whatever is noted down, make clear and intelligible, illustrating by a sketch if possible. Make the note so clear that reference to it after a long space of years would bring the whole subject before the mind in an instant. If this is not done the author of the note himself will not have patience to dig out the meaning when it is needed; and the note will be of no value.

METHOD OF DESIGN.

The fundamental lines of thought and action which every designer follows in the solution of any problem in any class of work whatsoever, are four in number. The expert may carry all these in mind at the same time, without definite separation into a a step-by-step process; but the student must master them in their proper sequence, and thoroughly understand their application. In these four are concentrated the entire art of Machine Design. When they have become so familiar as to be instinctively applied on any and all occasions, good design is the result. The only other quality which will facilitate still further the design of good machinery is experience; and that cannot be taught, it must be acquired by actual work.

1. **Analysis of Conditions and Forces.** First, take a good square look at the problem to be solved. Study it from all sides, view it in all lights, note the worst conditions which can possibly exist, note the average conditions of service, note any special or irregular service likely to be called for.

With these conditions well in mind, make a careful analysis of all the forces, maximum as well as average, which may be brought into play. Make a rough sketch of the piece under consideration, and put in these forces. Be sure that these forces are at least approximately right. Go over the analysis carefully again and again. Remember that time saved at the beginning by hasty and poor analysis will actually be time lost at the end; and if the machine actually fails from this reason, heavy financial loss in material and labor will occur. Any haste toward completion of the structure beyond the roughest outline, without this careful study of forces, is a blind leap in the dark, entirely unscientific, and almost certain to result in ultimate failure.

On the other hand this principle may be carried too far. In trying to make the analysis thorough and the forces accurate, it is quite possible to consume more than a reasonable amount of time. Again, it is not always easy, and frequently impossible, to determine exactly the forces acting on a given piece. But their *nature*, whether sudden or slowly applied, rapid in action or only occurring at intervals, and their *approximate* direction and magnitude at least, are always capable of analysis. There are few, if any, cases where close assumptions cannot be made on the above basis and the design proceeded with accordingly. Hence the danger of too great refinement of analysis is simply to be avoided by the designer's plain business sense.

The first tendency of the student is to pass over the study of the forces as dull and dry, and attempt the design at once. He soon finds himself facing problems of which he sees no possible solution, and he bases his design on pure guess-work. This is the only solution possible from such a point of view, and is really no solution at all. A guess which has some rational backing is often successful; but in that case some analysis is required, and it is not a pure guess, but falls under the very principle we are considering.

There is no short cut to the design of machine parts which avoids this full understanding of the forces that they must sustain. The size of a belt depends upon the maximum pull upon it, and the designing of belts is nothing but providing sufficient cross-section of leather to prevent the belt tearing under the pull. Again, if pulley arms are not to break, or shafts twist off, or bolts be torn apart, or the teeth of gears fail, or keys and pins shear off, we must first, of course, find out what forces exist which are likely to produce stress that may lead to such breakage. We should not guess at the sizes, and then run the machine to see if breakage results, and then guess again. Machines are sometimes built in this way, but it is an unreasonable and uncertain method. We must use every effort to foresee the stress which a piece is liable to receive, before we decide its size. We must know all the forces approximately, if not positively. The analysis must be thorough enough to permit of reasonable assumption, if not positive assertion. It is manifestly impossible to solve any problem until we know exactly what the problem is; and a full analysis *is* the statement of the problem.

2. **Theoretical Design.** After we know by careful analysis what stress the machine part has to sustain, the next step is so to design it that it will theoretically resist the applied forces with the least expenditure of material.

We often see machinery with the metal of which it is made distributed in the worst possible manner. In places where the stress is heavy and a rigid member is needed, we find a weak, springy part; while in other parts, where there are no forces to be resisted, or vibration to be absorbed, there seems to be a waste of good material. Whether in such case the analysis of the forces was poor, or perhaps not made at all, or whether a knowledge of how to design so as to resist the given forces was wholly absent, cannot be told. At any rate, lack of either or both is clearly shown in the result.

Any member of a machine may vary in form from a solid block or chunk of material to an open ribbed structure. The solid chunk fills the requirement as far as strength is concerned, unless it is so heavy as to fail from its own weight. But such construction is poor design, except in cases where the concentration of heavy mass is necessary to absorb repeated blows like those of a hammer. The possibility of these blows should, however, have been determined in the analysis; and the solid, anvil construction then becomes theoretical design for that analysis.

For steadily applied loads an open, ribbed, or hollow box structure can be made which will distribute the metal where it is theoretically needed, and each fiber will then sustain its proper share of the load. In this way weight, cost, and appearance are heeded; and the service of the piece is as good as, and probably better than, it would be with the clumsy, solid form.

There is no such thing as putting too much theory into the design of machinery. The strongest trait which an engineer can have is absolute faith in his analysis and calculations, and their reproduction in his theoretical design. Theoretical design is an indication of scientific advance in the art, and some of the greatest steps of progress which have been made in recent years have been accomplished through a purely theoretical study of machine structure.

It will never do, however, to be satisfied with theoretical design when it is not in accord with modern commercial and manufacturing considerations. Hence the next step after the determination of the theoretical design is the study of it from the producing standpoint.

3. Practical Modification. All theoretical design viewed from the business standpoint is worthless, unless it has been subjected to the test of cheap and efficient production. Each machine detail, though correct in theory, may yet be improperly shaped and unfit for the part it is to play in the general scheme of manufacture.

The conditions here involved are changeable. What is good design in this decade may be bad in the next. In this light the designer must be a close student of the signs of the times; he must follow the march of progress, closely applying existing resources, conditions, and facilities, otherwise he cannot produce up-to-date designs. The introduction of new raw materials, the cheapening of production of others, the changing of shop methods, the use of special machinery, the opening of new markets, the development of new motive agents,—all these and many others are constantly demanding some modification in design to meet competition.

Illustrative of this, note the change which has been wrought by the development of electric power, the rise and decline of the bicycle business, the present manufacture of automobiles, the last named especially with reference to the development of the small motive unit, the gasolene engine, the steam engine, etc. The design of much machinery has been materially changed to meet the exacting demands of these new enterprises.

Practical modifications of design necessary to meet the limitations of construction in the pattern shop, foundry, and machine shop are of daily application in the designer's work. He must keep in his mind's eye at all times the workmen and the processes they use to create his designs in metal in the shop.

"How can this be made?" "Can it be made at all?" "Can it be made cheaply?" "Will it be simple in operation after it is made?" "Can it be readily removed for repair?" "Can it be lubricated?" "How can it be put in place?" "How can it be gotten out?" "Will it be made in small quantities or large?" "Will it sell as a special or standard machine?" etc., etc.

The consideration of such questions as these is a practical necessity as a business matter. No other feature affects the design of machinery more, perhaps; for designs which cannot be built as business propositions are no designs at all.

The student, it is true, may not have the extended shop knowledge which is essential to this; but he can do much for himself by visiting shops whenever possible, getting hold of shop ways of doing things, and invariably treating his work as a business matter. Though a man may not be a pattern maker, molder, blacksmith, or machinist, yet he can soon gain ideas of the processes in each of these branches which will be of immense advantage to him in his designing work.

4. Delineation and Specification. This means the clear and concise representation of the design by mechanical drawings.

This is as much a part of the routine method of Machine Design as the other three points which have been discussed. The mere act of putting the results of mechanical thinking on paper is one of the greatest helps to force thinking machinery to systematic and definite action. A designer never thinks very long without drawing something, and the student must bring himself to feel that a drawing in its first sense is a means of helping his own thought, and must freely use it as such.

In its second and final sense, the drawing is an order and specification sheet from the designer to the workman. Design which stops short of exact, finished delineation in the form of working shop drawings is only half done. In fact the possibility of a piece being thus exactly drawn is often the crucial test of its feasibility as a part of a machine. It is easy to make general outlines, but it is not so easy to get down to finished detail. It is safe to say that there is no one thing productive of more trouble, delay and embarrassment, and waste of time and money in the shop, when there need be none from this cause, than a poor detail drawing. The efficiency of the process of design is not fully realized, and failures are often recorded where there should be success, merely because the indefiniteness permitted by the designer in the drawings naturally transmitted itself to the workman, and he in turn produced a part indefinite in form and operation.

The actual process of drawing in the development of a design may be outlined as follows:

Rough sketches merely representing ideas, not drawn to scale, are first made. These are of use only so far as the choice of mechanical ideas is concerned, and to carry preliminary dimensions.

Following these sketches, comes a layout to scale, of the favored sketch, a working out of the relative sizes and location of the parts. This drawing may be of a sketchy nature, carrying a principal dimension here and there to fix and control the detailed design. In this drawing the design is developed and general detail worked out. The minute detail of the individual parts is, however, left to the subsequent working drawing.

This layout drawing may now be turned over to an expert draftsman or detail designer, who picks out each part, makes an exact drawing of it, studying every little detail of its shape, and finally adds complete dimensions and specifications so that the workman is positively informed as to every point of its construction.

General drawings and cross sections constitute the last step in the process of complete delineation. These show the parts assembled in the complete machine. They also serve a valuable purpose to the draftsman in checking up the dimensions of the detail drawings. Errors which have escaped previous notice are often discovered in this way. The layout, mentioned above, is sometimes finished up into a general drawing; but it is safer to make an entirely new drawing, as changes in detail are often necessary after the layout is made.

The four fundamental lines of thought and action noted above may be summarized thus—**"analyze and theorize, modify and delineate."** This is a maxim easy to remember, applicable to every problem in Machine Design, and always provides the answer to the question "What shall I do, how shall I proceed?" by pointing out the proper sequence in the course to be followed.

CONSTRUCTIVE MECHANICS.

Mechanics is a constructive science, its principles lying at the root of the design and operation of all machinery. It is usually taught, however, as an advanced mathematical subject; and the student gets his original conceptions of forces, moments, and beams in the abstract, before he realizes the constructive value of such conceptions. By "Constructive Mechanics" is meant the study of a machine purely from its constructive side, the viewing of the parts with respect to their "mechanics," and satisfying the requirements of the same in form and arrangement.

The student may cultivate this habit of clear, mechanical perception by constantly noting the "mechanics" of the simple structures which he sees in his daily routine of work. Aside from machinery, in which the "mechanics" is often obscure, the world is full of simple examples of natural strength and symmetry, explainable by application of the principles of pure "mechanics."

Posts and pillars are largest at their bases; overhanging brackets or arms are spread out at the fastening to the wall; heavy swinging gates are counter-balanced by a ponderous weight; the old-fashioned well sweep carries its tray of stones at the end, adjusting the balance to a nicety; these are examples of things depending for their form and operation upon the principles of "mechanics." The building of them involved "constructive mechanics," and yet their constructor perhaps never heard of the science, using merely his natural sense of mechanical fitness Such simple reasoning is, however, Constructive Mechanics.

Forces, Moments, and Beams. Machines are nothing but a collection of (1) parts taking direct stress, or (2) parts acting as loaded beams. Forces acting *without* leverage produce direct stress on the sustaining part. Forces acting *with* leverage produce a moment; the sustaining member is a beam, and the stress therein depends on the theory of beams, as explained in "Mechanics."

An example of the first is the load on a rope, the force acting without leverage, and the rope therefore having a direct stress put upon it.

An example of the second is a push of the hand on the crank of a grindstone. A moment is produced about the hub of the crank; the arm of the crank is a beam, and the stress at any point of it may be found by the method of theory of beams.

Tension, Compression, and Torsion. The stress induced in the sustaining part, whether tensile, compressive, or torsional, is caused by the application of forces, either acting directly without leverage, or with leverage in the production of moments.

The forces applied from external sources are at constant war with the resisting forces due to the strength of the fibres of the material composing the machine members. The moments of the *external* forces are constantly exerted against and balanced by the moments of the *internal* resistance of the material. Hence, design, from a strength standpoint, is merely a balancing of internal strength against external force. In other words, we may in all cases write a sign of equality, place the applied effort on one side, the effective resistance on the other, and we shall have an equation, which, if capable of solution, will give the proper proportions of the parts considered.

External Force = Internal Resistance.

External Moment = Internal Moment of Resistance.

Expressed in terms of the "Mechanics:"

$$P=AS \qquad (1)$$

$$B \text{ or } T=\frac{SI}{c} \qquad (2)$$

In these formulas, which are perfectly general,

P=direct load in pounds.

A=area of effective material, in square inches.

S=working fibre stress of the material (tensile, compressive, or shearing), in pounds per square inch.

B or T=external moment (bending or torsional), in inch-pounds.

I=moment of inertia (direct or polar), of the resisting section.

c=distance of the most remote fibre of the resisting section from the neutral axis.

P may produce direct tensile, compressive, or shearing stress.

B may produce tensile or compressive stress, and requires use of direct moment of inertia in either case.

T produces shearing stress, and requires use of polar moment of inertia.

The origin of formula (1) is obvious, the assumption being that the fibre stress is equally distributed to every particle in the area "A."

The development of formula (2) is given in any text-book in Mechanics. It requires the aid of the Calculus, however. Any good handbook gives values for both the direct moment of inertia and the polar moment of inertia for quite a large variety of sections, so that further reference is an easy matter for the student. These values are also obtained through the methods of the Calculus.

The reason for introducing these formulas at this time is to call the attention of the student especially to the fact of their universal and fundamental use in all problems concerning the strength of machine parts. Nearly every computation may be reduced to or expanded from these two simple equations. Many complex combinations occur, of course, which will not permit simple and direct application of these formulas, but the student will do well to place himself in perfect command of these two. Assuming that he is able to analyze forces, and compute the simple moment at the point where he wishes to find the strength of section, the rest is the mere insertion of the assumed working fibre stress of the material in the formula (2) above, and solution for the quantity desired.

When the case is one of combined stress, the relation becomes more complicated and difficult of analysis and solution. The most common case is where bending is combined with torsion, as in the case of a shaft transmitting power, and at the same time loaded transversely between bearings. In fact there are very few cases of shafts in machines, which, at some part of their length, do not have this combined stress. In this case the method of procedure is to find the simple bending moment and the simple torsional moment separately, in the ordinary way. Then the theory of elasticity furnishes us with a formula for an equivalent bending or an equivalent torsional moment which is supposed to produce the same effect upon the fibres of the material as the combined action of the two simple moments acting together. In other words, the separate moments combined in action, being impossible of solution in that form, are reduced to an equivalent simple moment and the solution then becomes the same as for the previous case.

These equivalent equations are given below, the subscript "e" being added to express separation from the simple moment:

$$B_e = \frac{B}{2} + \tfrac{1}{2}\sqrt{B^2+T^2} \qquad (3)$$

$$T_e = B + \sqrt{B^2+T^2} \qquad (4)$$

B_e and T_e, found from these equations, are the external moments, and are to be equated to the internal moments of resistance of the section precisely as if they were simple bending or torsional moments. Either may be used. For shafts (4) is generally used, being the simpler of the two in form.

FRICTION AND LUBRICATION.

The parts of a machine which have no relative motion with regard to each other are not dependent upon lubrication of their surfaces for the proper performance of their functions. In cases where relative motion does occur, as between a planer bed and its ways, a shaft and its bearing, or a driving screw and its nut, friction, and consequent resistance to motion, will inevitably occur. Heat will be generated, and cutting or scoring of the surfaces will take place if the surfaces are allowed to run together dry.

This difficulty, which exists with all materials, cannot be overcome, for it is a result of roughness of surface, characteristic of the material even when highly finished. The problem of the designer, then, is to take conditions as he finds them, and, as he cannot change the physical characteristics of materials, so choose those which are to rub together in the operation of the machine that friction will be reduced to the lowest possible limit. Now it fortunately happens that there are certain agents like oil and graphite, which seem to fill up the hollows in the surface of a solid material, and which themselves have very little friction on other substances. Hence, if a machine permits by its design an automatic supply of these lubricating agents to all surfaces having motion between them, friction may be reduced to the lowest limit.

If this full supply of lubricant be secured, and the parts still heat and cut, then the fault may be traced to other causes, such as springy surfaces, localization of pressure, or insufficient radiating surface to carry away the heat of friction as fast as it is generated.

Lubricating agents are of a nature running from the solid graphite form to a thick grease, then to a heavy dark oil, and finally to a thin, fluid oil flowing as freely as water. The solid and heavy lubricants are applicable to heavily loaded places where the pressure would squeeze out the lighter oils. Grease, forced between the surfaces by compression grease cups, is an admirable lubricator for heavy machinery under severe service. High-speed and accurate machinery, lightly loaded, requires a thin oil, as the fits would not allow room for the heavier lubricants to find their way to the desired spot. The ideal condition in any case is to have a film of lubricant always between the surfaces in contact, and it is this condition at which the designer is always aiming in his lubricating devices.

Oil ways and channels should be direct, ample in size, readily accessible for cleaning, and distributing the oil by natural flow over the full extent of the surface. Hidden and remote bearings must be reached by pipes, the mouths of which should be clearly indicated and accessible to the operator of the machine. Such pipes must be straight, if possible, and readily cleaned.

There is one practical principle affecting the design of methods of lubrication of a machine which should be borne in mind. This is, "Neglect and carelessness by the operator *must* be provided for." It is of no use to say that the ruination of a surface or hidden bearing is due to neglect by the operator, if the means for such lubrication are not perfectly obvious. This is "locking the door after the horse is stolen." The designer has not done his duty until he has made the scheme of lubrication so plain that every part *must* receive its proper supply of oil, except by gross and willful negligence, for which there can be no possible just excuse.

WORKING STRESSES AND STRAINS.

Some persons object to the use of these terms, as one is frequently used for the other, and misunderstanding results. This is doubtless true; but the student may as well learn the true relation of the terms once for all, because he will frequently run across them in his reading and reference work, and should interpret them rightly. The strict relation of the two is as follows:

Stress is the internal force in a piece resisting the external force applied to it. A weight of ten pounds hanging on a rope produces a *stress* of ten pounds in the rope.

Strain is the change of shape, or deformation, in a piece resisting an external force applied to it. If the above weight of ten pounds stretches the rope $\frac{1}{4}$ inch, the *strain* is $\frac{1}{4}$ inch.

Unit stress is stress per unit area, e. g., per square inch.

Unit strain is strain per unit length, e. g., per inch length.

In the above case, if the rope were $\frac{1}{2}$ square inch in area and 30 inches long, the unit stress, or intensity of stress, is $10 \div \frac{1}{2} = 20$ pounds per square inch; the unit strain is $\frac{1}{4} \div 30 = \frac{1}{120}$ inch per inch.

When stress is induced in a piece, the strain is practically proportional to the stress for all values of the stress below the elastic limit of the material; and when the external load is removed the strain will entirely disappear, or the recovering power of the material will restore the piece to the original length.

Illustrating by the case above, on the supposition that the elastic limit has not been reached by the stress of 20 pounds per square inch, if the load of 10 pounds were taken off, the $\frac{1}{4}$-inch strain would disappear and the rope return to its original length; if the load were changed to $\frac{1}{2}$ of 10 pounds, or 5 pounds, the strain would be $\frac{1}{2}$ of $\frac{1}{4}$ inch, or $\frac{1}{8}$ inch.

Now it is found that if we wish a piece to last in service for a long time without danger of breakage, we must not permit it to be stressed anywhere near the elastic limit value. If we do, although it will probably not break at once, it is in a dangerous condition, and not well suited to its requirements as a machine member. The technical name for this weakening effect is "fatigue." It is further found that the fatigue due to this repeated stress is reached at a lower limit when the stress is alternating in character than when it is not. In other words, if we first pull on a piece and then push on it, we shall first have the piece in tension and then in compression; this alternation of stress repeated to near the elastic limit of the material will fatigue it, or wear out the fibres, and it will finally fail. If, however, we first pull on the piece with the same force as before, and then let go, we shall first have the piece in tension and then entirely relieved; such repetition of stress will finally "fatigue" the material, but not so quickly as in the first case. Experiments indicate that it may take twice as many applications in the latter case as in the former.

The working stress of materials permissible in machines is based on the above facts. The breaking strength divided by a liberal factor of safety will not necessarily give a desirable working stress. The question to be answered is, "Will the assumed working fibre stress permit an indefinite number of applications of the load without fatiguing the material?"

Hence we see that the same material may be safely used under different assumptions of working stress. For example, a rotating shaft, heavily loaded between bearings. acts as a beam which in each revolution is having its particles subjected, first to a maximum tensile stress, and then to a maximum compressive stress. This is obviously a very different stress from that which the same piece would receive if it were a pin in a bridge truss. In the former we have a case where the stress on each particle reverses at each revolution, while in the latter we have merely the same stress recurring at intervals, but never becoming of the opposite character. For ordinary steel, a value of 8,000 would be reasonable in the former case, while in the latter it may be much higher with safety, perhaps nearly double.

From the facts stated above, it is evident that exact values for working fibre stress cannot be assumed with certainty and applied broadly in all cases. If the elastic limit of the material is definitely known we can base our working value quite surely on that.

With but a general knowledge of the elastic limit, ordinary steel is good for from 12,000 to 15,000 pounds per square inch non-reversing stress, and 8,000 to 10,000 reversing stress. Cast iron is such an uncertain metal on account of its variable structure that stresses are always kept low, say from 3,000 to 4,000 for non-reversing stress, and 1,500 to 2,500 for reversing stress.

With these values as a guide, and the special conditions controlling each case carefully studied, reasonable limits may be assigned for working stress, not only of steels, various grades of cast iron, and mixtures of the same, but of other alloys, brass, bronze, etc. Gun metal, semi-steel, and bronze are intermediate in strength between cast iron and steel. Data on the strength of materials are available in any of the handbooks, and should be consulted freely by the student. They will be found somewhat conflicting, but will assist the judgment in coming to a conclusion.

Application to Practical Case. In actual practice the only information which the designer has, upon which to base his design, is the object to be accomplished. He must choose or originate suitable devices, develop the arrangement of the parts, make his own assumptions regarding the operation of the machine, then *Analyze and Theorize*, *Modify and Delineate* each detail as he meets it.

This, it will be found, is a very different matter from taking some familiar piece of machinery, such as a pulley, or a shaft, or a gear, as an isolated case, the load being definitely given, and proceeding with the design. This is easily done, but is only half the problem, for machine parts, such as pulleys, gears, and shafts, do not confront the designer tagged or labeled with the conditions they are to meet. He is to provide parts to meet the specific conditions, and it is as much a part of his designing method to know how to attack the design of a machine as it is to know how to design the parts in detail after the attack has reduced the members to definitely loaded structures. The whole process must be gone through, the preliminary sketches, calculations, and layout, all of which precede the detail design and working drawings; and no step of the process can be omitted.

It is for this reason that the present case used for illustration is carried out quite thoroughly. The student should make himself familiar with every step of the designing method as applied to this simple case of design. More complex problems, handled in the same way, will simplify themselves; and when the point is reached where confidence exists to take hold of the design of any machine, however unfamiliar its object may be, or however involved its probable detail appears, the student has become the true designer. It is the knowing how to attack a problem, to start definite work on it, to go ahead boldly, confident that the method applied will produce results, that gives command of the design of machinery and wins engineering success.

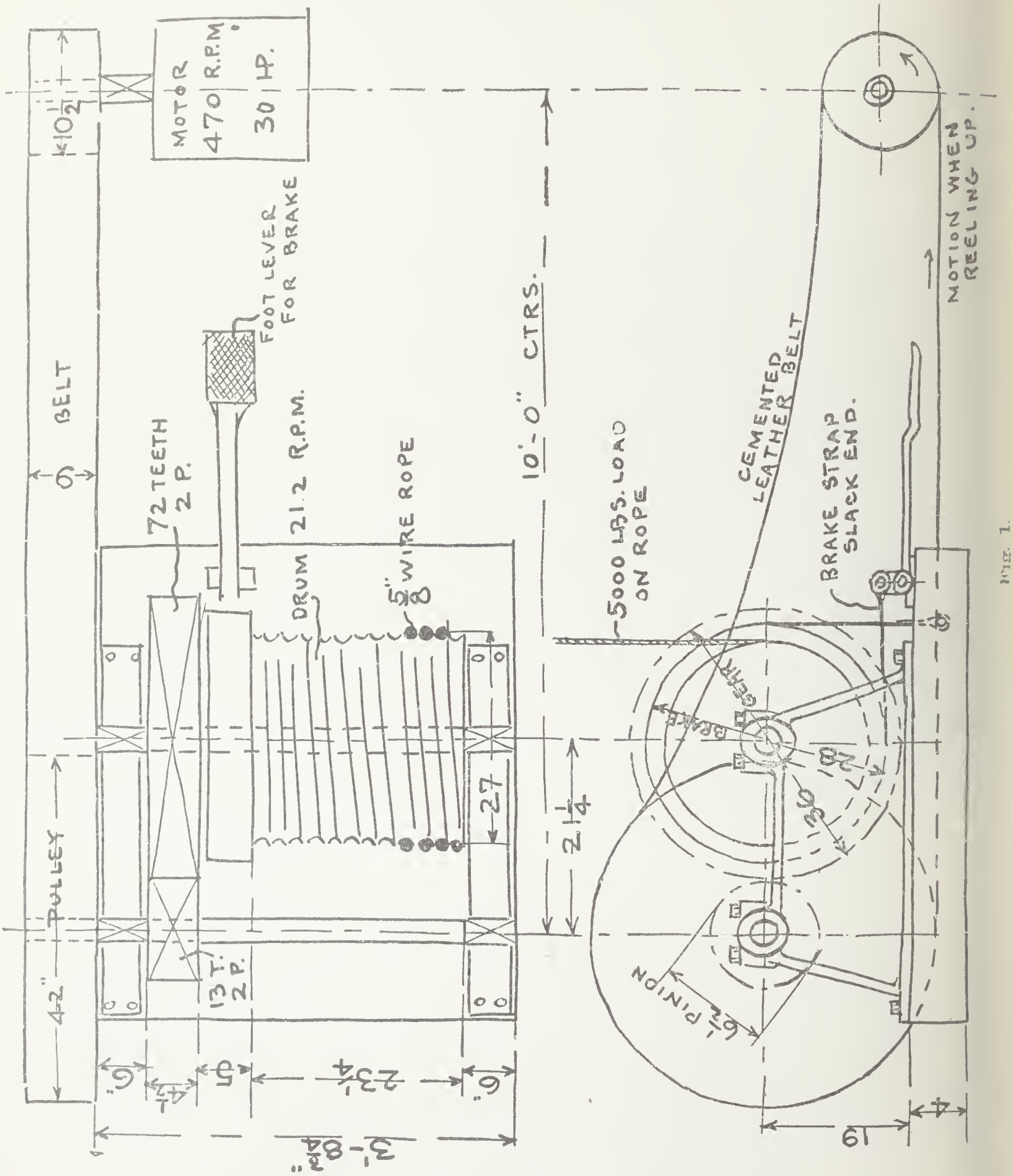


Fig. 1.

The special case which has been chosen to illustrate the application of the principles stated in the foregoing pages is ideal, in that it does not represent any actual machine at present in operation. Probably builders of hoisting machinery have devices which would improve the machine as shown. In detail, as well as arrangement, they could doubtless make criticism as manufacturers. The arrangement as shown is merely intended to bring out in simplest form the common elements of transmission machinery as parts of some definite machine, instead of as isolated details. The design is one entirely possible, practical, and mechanical, but special attention has been paid to simplicity in order to enable the student to follow the method closely, for the *method* is the chief thing for him to acquire.

The student is expected to refer constantly to Part II for a more formal and general discussion of the simple machine elements involved in the case considered. Part II is intended to be a simplified and condensed reference book, carried out in accordance with the method of machine design as specified in Part I. The student should not wait until he has completed the study of this part before taking up Part II, for the latter is intended for use with the former in the solution of the problems.

In the case of power transmission about to be studied, the running, conversational method employed assumes that the student is in possession of the matter in Part II on the subject considered. Thus, in the design of the pulley, reference to the subject of "Pulleys" in Part II is necessary to follow the train of calculation; in designing the gear, consult "Gears;" in calculating size of shafts, see "Shafts," etc., etc.

Problem. A machine is to be designed to be set on the floor of a building to drive a wire rope falling from the overhead sheaves of an elevator or hoist. Without regard to details of this overhead arrangement, for its design would be a separate problem, suppose that the data for the rope are as follows:

Load on rope	5,000 pounds.
Speed of rope	150 feet per minute.
Length of rope to be reeled in	200 feet.

We shall further assume that the driving power is to be an electric motor belted to the machine, that the required speed reduction can be satisfactorily obtained by a single pair of pulleys and one pair of gears, and that a plain band brake is to be applied to the drum.

With this data we shall proceed to work out the detail design of the machine.

Preliminary Sketch. The first thing to do is to sketch roughly the proposed arrangement of the machine.

This might appear like Fig. 1 except that it would have no dimensions in addition to the data given above. If the scheme seems suitable, the next step is to make such preliminary calculations as will give further data, exact or closely approximate sizes, to be put at once on the sketch, to outline the future design.

Rope and Drum. Referring to tables of strength of wire rope (Kent's Pocket Book gives the manufacturers' list), we find that a $\frac{5}{8}$-inch cast-steel rope will carry 5,000 pounds safely, and that the proper size of drum to avoid excessive bending of the rope around it is 27 inches diameter.

Allowing $\frac{1}{8}$ inch between the coils as the rope winds on the drum, the pitch of coil will be $\frac{3}{4}$ inch as shown in sketch, Fig. 2. The length of one complete coil is, practically, $\frac{27\times 3.1416}{12}$ $=7.07$ feet. To provide for 200 feet will require $\frac{200}{7.07}=28+$coils. To be safe, let us provide for 30 coils, for which a length of drum $(30\times\frac{3}{4})+\frac{3}{4}=23\frac{1}{4}$ inches is required.

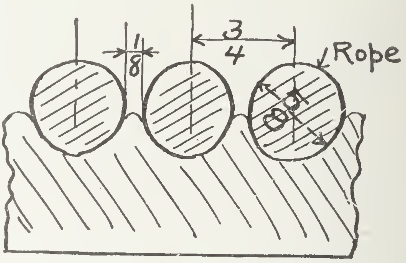


Fig. 2.

The space for brake strap may be assumed at 5 inches, and the thickness to provide necessary strength determined later in the design. The frictional surface of the strap may be of basswood blocks, say $1\frac{1}{4}$ inches thick, screwed to the metal band. The diameter of brake surface may be 28 inches.

Driving Gears. The size of drum gear evidently depends upon the method of fastening to the drum, and, other things being equal, should be kept as small as possible. One way would be to key the gear on the outside of the drum, another to bolt the gear to the end of the drum. The latter has the advantage that a standard gear pattern can be used with the slight change of addition of bolt flange on the arms. This makes a simple, direct, and strong drive, the bolts being in shear.

Sketching this arrangement as the preferred one (Fig. 2A), it is evident that the diameter of the gear should be at least as large as the drum in order to keep the tooth load down to a reasonable figure. On the other hand, if made too large, it spreads out the machine and destroys its compactness. As a diameter of 36 inches is not excessive, let us assume this, and see if a desirable proportion of gear tooth can be found to carry the load.

For a pitch diameter of 36 inches there will be a theoretical load of $\frac{5,000 \times 27}{36} = 3,750$ pounds at the pitch line. But the load on the tooth must not only impart a pull of 5,000 pounds to the rope, but must overcome friction between the gear teeth in action, also between the drum shaft and its bearings. Assuming the efficiency between the rope and tooth load to be 95 per cent, the net load, therefore, which the tooth must take is $\frac{3,750}{.95} = 3,947$, say 4,000 pounds.

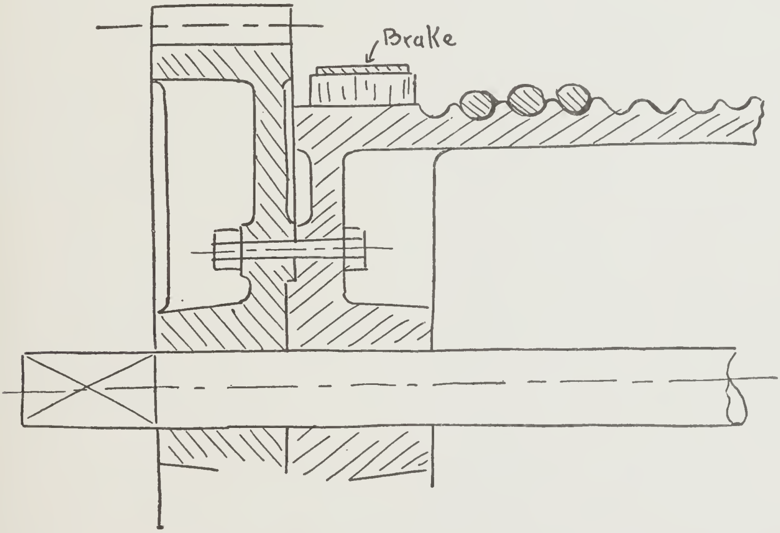


Fig. 2A.

Assuming involute teeth, and applying the "Lewis" formula, (Part II, "Gears"):

$$W = s \times p \times f \times y$$

$$4{,}000 = 6{,}000 \times p \times f \times .116$$

$$p \times f = \frac{4{,}000}{6{,}000 \times .116} = 5.7 \text{ inches}$$

$W = 4{,}000$
$s = 6{,}000$
$y = .116$ (number of teeth assumed at 75)
$p =$ circular pitch
$f =$ face of gear

Let $f = 3\,p$ (a reasonable proportion for machine-cut teeth).

Then $3 \times p^2 = 5.7$

$$p^2 = 1.9$$

$$p = \sqrt{1.9} = 1.378 \text{ inches}$$

The diametral pitch corresponding to this is

$$\frac{3.1416}{1.378} = 2.28$$

which is just between the regular standard pitches, 2 and $2\frac{1}{2}$, for which stock cutters are made. To be safe, let us take the coarser pitch, which is 2. The circular pitch corresponding to this is $\frac{3.1416}{2} = 1.57$, and making the face about three times the circular pitch gives

$$3 \times 1.57 = 4.71, \text{ say } 4\tfrac{1}{2} \text{ inches.}$$

The number of teeth in the gear is then $36 \times 2 = 72$. Referring to the value assumed for the tooth factor in calculation above, it is seen that y was based on 75 as the number of teeth, which is near enough to 72 to avoid the necessity of further checking the result.

The pinion to mesh with this gear should be as small as possible in order to get a high-speed ratio between pinion shaft and drum, otherwise an excessive ratio will be required in the pulleys, making the large one of inconvenient size. Small pinions have the teeth badly undercut and therefore weak, 13 teeth being the lowest limit usually considered desirable, for that reason. Choosing that number, we have a pitch diameter of $\frac{13}{2} = 6.5$ in., which is probably ample to take the shaft and key, and still leave sufficient stock under the tooth for strength. If made of cast iron, however, the pinion teeth, on account of the low number, will be narrower at the root than those of the gear of 72 teeth. Yet it was upon the basis of the latter that the pitch was chosen, for it will be remembered that the value of y in the formula was taken at .116. Hence the pinion will be weaker than the gear unless we make it of stronger material than cast iron, of which the large gear is supposed to be made. Steel lends itself very readily to this requirement; and in practice, pinions of less than 20 teeth are usually made of this material, hence we shall specify the pinion to be of steel.

Pulleys. The question now is whether or not we can get a suitable ratio in the pulleys without making the large one of inconvenient size, or giving the motor too slow speed for an economical proportion.

Suppose we limit ourselves to a diameter of 42 inches for the large pulley, and try a ratio of 4 to 1; this will give a diameter for the small pulley of $\frac{42}{4}=10\frac{1}{2}$ inches. We shall then have

Total ratio between drum and motor........ $\frac{72}{13}\times 4=\frac{288}{13}=22.2$

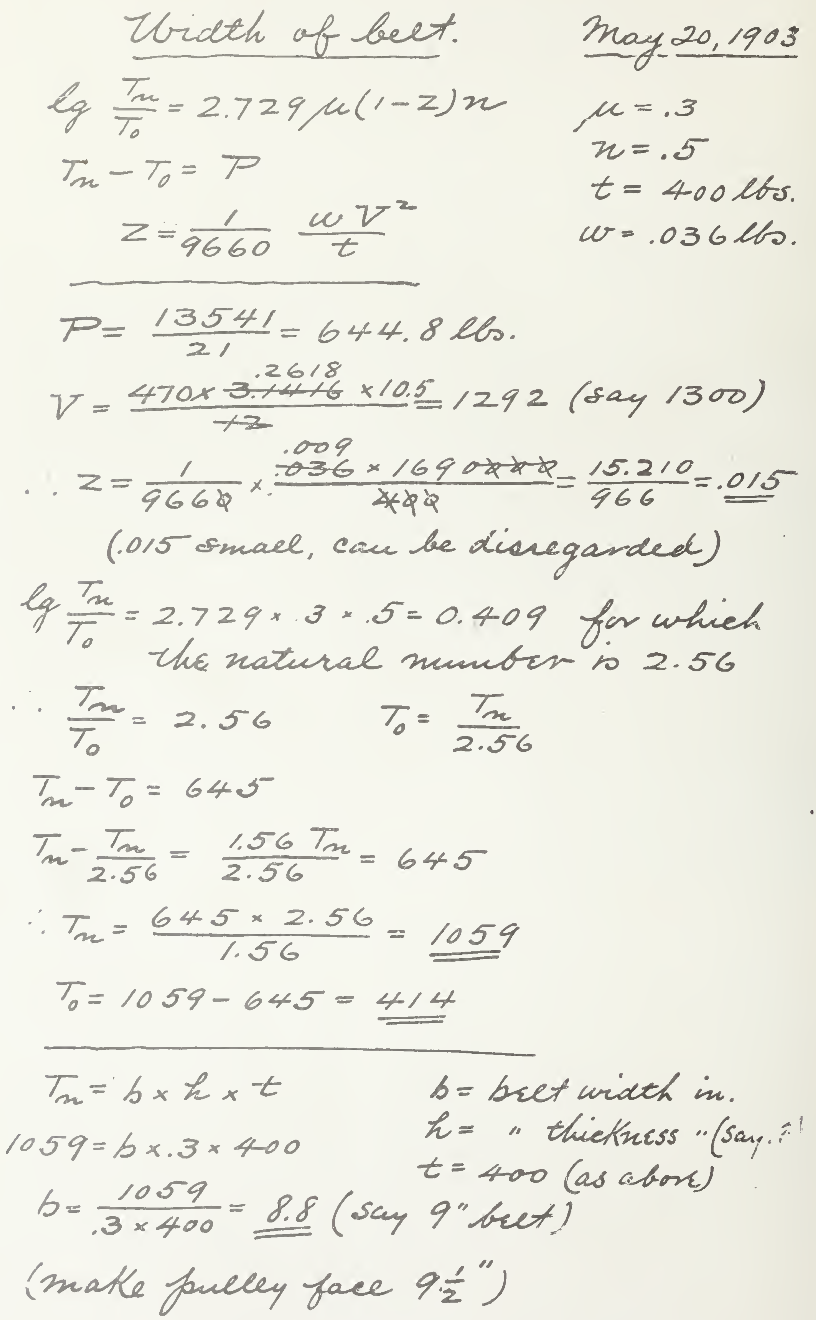
Rev. per min. of drum to give 150 f. p. m. of rope .... $\frac{150}{7.07}=21.2$

Rev. per min. of motor........................$22.2\times 21.2=470$

Horse-power of motor at 80 per cent efficiency $\frac{150\times 5{,}000}{33{,}000\times .80}=30$

A 30 H. P. motor running 470 r. p. m. would be classed as a slow speed motor and would be a heavier machine and cost more than one of higher speed. It will be noticed, however, that the diameter of the small pulley is already quite reduced, and it is hardly desirable to decrease it still further. Neither can we increase the large pulley, as we have already set the limit at 42 inches. Hence, for our present problem we cannot improve matters much without increasing the size of the large gear, which is undesirable, or putting in another pair of gears, which is contrary to the conditions of the problem. As such a motor is perfectly reasonable, we shall assume it to be chosen for the purpose.

In commercial practice it would be well to pick out some standard make of motor of the required horse-power, note the speed as specified by the makers, and then, if possible, suit the ratio in the machine to this speed. It is always best to use standard machinery, if possible, both from the standpoint of first cost, as well



Width of belt. May 20, 1903

$$\lg \frac{T_n}{T_0} = 2.729\,\mu(1-z)n$$

$$T_n - T_0 = P$$

$$z = \frac{1}{9660}\,\frac{wV^2}{t}$$

$\mu = .3$

$n = .5$

$t = 400$ lbs.

$w = .036$ lbs.

$$P = \frac{13541}{21} = 644.8 \text{ lbs.}$$

$$V = \frac{470 \times \overset{.2618}{\cancel{3.1416}} \times 10.5}{\cancel{12}} = 1292 \text{ (say 1300)}$$

$$\therefore z = \frac{1}{966\cancel{0}} \times \frac{\overset{.009}{\cancel{.036}} \times 169\cancel{0000}}{\cancel{400}} = \frac{15.210}{966} = \underline{\underline{.015}}$$

(.015 small, can be disregarded)

$$\lg \frac{T_n}{T_0} = 2.729 \times .3 \times .5 = 0.409$$ for which the natural number is 2.56

$$\therefore \frac{T_n}{T_0} = 2.56 \qquad T_0 = \frac{T_n}{2.56}$$

$$T_n - T_0 = 645$$

$$T_n - \frac{T_n}{2.56} = \frac{1.56\,T_n}{2.56} = 645$$

$$\therefore T_n = \frac{645 \times 2.56}{1.56} = \underline{\underline{1059}}$$

$$T_0 = 1059 - 645 = \underline{\underline{414}}$$

$$T_n = b \times h \times t$$

$$1059 = b \times .3 \times 400$$

b = belt width in.

h = " thickness " (say.

t = 400 (as above)

$$b = \frac{1059}{.3 \times 400} = \underline{\underline{8.8}} \text{ (say 9" belt)}$$

(make pulley face $9\frac{1}{2}$")

Fig. 3.

as ease of replacing worn parts. Machinery ordered special is expensive in first cost of designing, patterns, and tools, and extra spare parts for emergency orders are not often kept on hand.

Tabulation of Torsional Moments. For future reference, it is desirable at this point to tabulate the torsional moment, or torque, about each of the three shaft axes, assuming reasonable efficiencies for the various parts, as follows:

Efficiency between drum and gear tooth	95 per cent
Efficiency between drum and pinion shaft	90 per cent
Efficiency between drum and motor shaft	80 per cent

TABLE OF TORSIONAL MOMENTS.

Axis.	Inch Lbs. Torque at 100 Per Cent Efficiency.	Inch Lbs. Torque, Efficiency as Above.
Drum	$5{,}000\times\frac{27}{2}$ .... $=67{,}500$	$\frac{67{,}500}{.95}=71{,}052$
Pinion	$5{,}000\times\frac{27}{2}\times\frac{13}{72}$ ...... $=12{,}187$	$\frac{12{,}187}{.90}=13{,}541$
Motor	$5{,}000\times\frac{27}{2}\times\frac{13}{72}\times\frac{10.5}{42}=3{,}047$	$\frac{3{,}047}{.80}=3{,}809$

This means that the motor develops a torque of 3,809 inch-pounds delivering to pinion shaft 13,541 inch-pounds, and to drum 71,052 inch-pounds.

Width of Belt. The page of calculation for belt width is reproduced in Fig. 3.

The calculation as given is strictly scientific, based on the working strength of a cemented joint ($t=400$ lbs. per square inch). This is a favorable situation for the use of a cemented joint, because it is easy to provide means of adjusting the belt tension by placing the motor on a sliding base. Otherwise a laced joint could be used, requiring relacing when the belt slackens through its stretch in service. Under the assumption that a double *laced* belt is used, the empirical formula below is one often applied:

$$\text{H. P.}=\frac{w\times V}{540}=\frac{w\times 1{,}300}{540}=30$$

$$\text{This gives } w=\frac{540\times 30}{1{,}300}=12{\cdot}4 \text{ inches (say 12 inches).}$$

It should be remembered that this value is purely empirical; it applies to a *laced* joint, and could not be expected to check the value of 9 inches obtained by the first computation for a cemented joint. It is fairly in proportion. For the quite definite service required of the belt in the present case, the width of 9 inches is doubtless sufficient, considering the cemented joint.

Length of Bearings. Considerable latitude in choice of length of bearings is permissible, especially in such slow-speed machinery. There is probably little danger from heating, and the question then becomes one of wear. It is better in such cases as the one in question, to choose boldly a length which seems to be reasonable and proceed with the design on that basis, even if the length be later found out of proportion to the shaft diameter, than to waste too much time in the preliminary calculation over the exact determination of this question. Probably in most cases of commercial practice the existence of patterns, or some other practical consideration, will decide the limits of length.

In the present instance it seems reasonable that a length of 6 inches would fill the requirement for the worst case, that of the drum shaft, and it is obvious that the bearings for the pinion shaft would naturally be of the same length on account of being cast on the same bracket, and faced at the same setting of the planer tool.

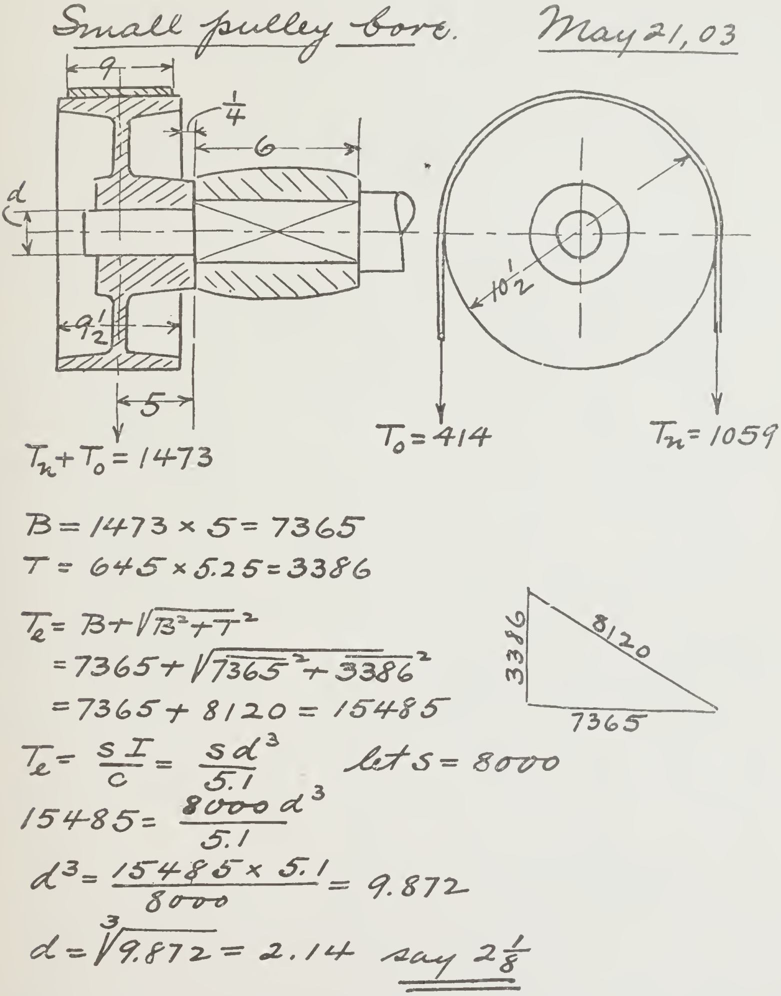
Height of Centers. The large pulley should naturally swing clear of the floor. This will require, say, a total height of 23 inches, out which we may take 4 inches for the base, leaving 19 inches as the height, center of bearing to base of bracket.

Data on Sketch. The data as found above should now be put on the sketch previously made; it will then have the appearance shown in Fig. 1.

This sketch is now in form to control all the subsequent detail design, and it is expected that the figured dimensions as shown can be maintained. It is impossible to predict this with positiveness, however, as in the working out of the minor details certain changes may be found desirable, when, of course, they should be made.

The shaft sizes do not appear on this sketch, hence before proceeding further the several shaft diameters must be calculated.

Sizes of Shafts. The calculations of the shaft diameters are good instances of systematic engineering computations, hence they are reproduced in the exact form in which they were made. The student should learn a valuable lesson in making and recording calculations by following these carefully. Note that each set of figures is independent, both in the statement of given data, as well as in the actual computation. Observe how easy it would be for the author of these figures or anyone else to check them even after a long lapse of time. If the machine should unexpectedly fail in service the figures are always available to prove or disprove theoretical weakness. The right triangles merely indicate that the value of $\sqrt{B^2+T^2}$ was found by the graphical method suggested in



Small pulley bore. May 21, 03

$B = 1473 \times 5 = 7365$

$T = 645 \times 5.25 = 3386$

$$T_e = B + \sqrt{B^2+T^2}$$
$$= 7365 + \sqrt{7365^2 + 3386^2}$$
$$= 7365 + 8120 = 15485$$

$T_e = \frac{sI}{c} = \frac{sd^3}{5.1}$ let $s = 8000$

$$15485 = \frac{8000\,d^3}{5.1}$$

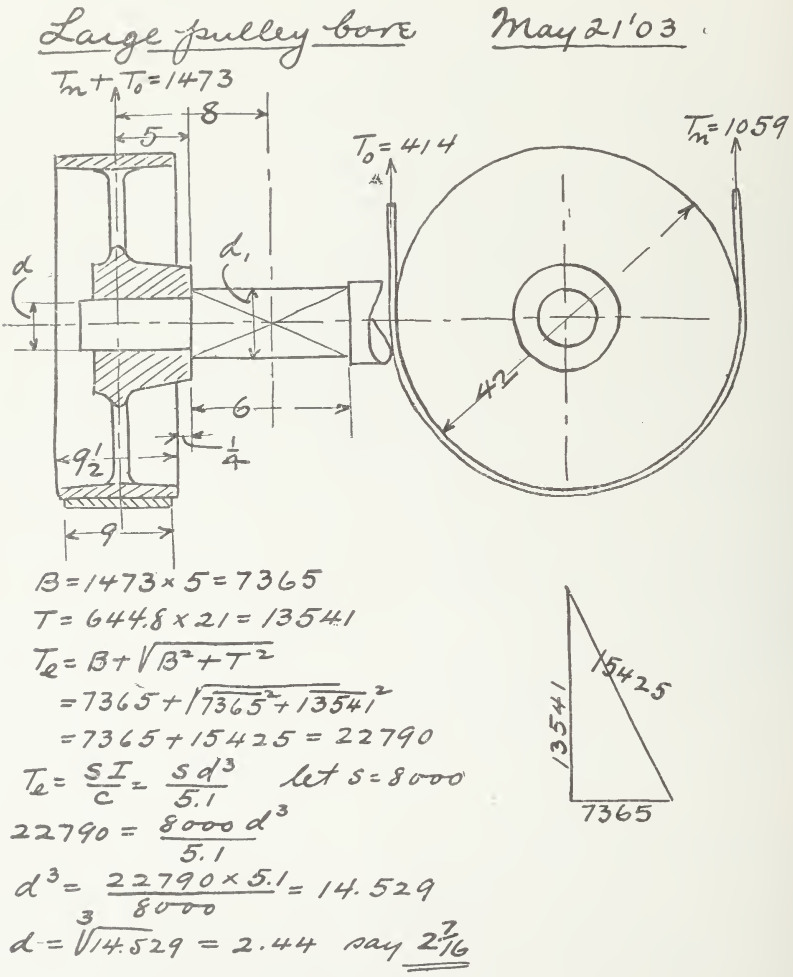
$$d^3 = \frac{15485 \times 5.1}{8000} = 9.872$$

$$d = \sqrt[3]{9.872} = 2.14 \quad \text{say } \underline{\underline{2\tfrac{1}{8}}}$$

Fig. 4.

Part II, "Shafts," the figures being put on the triangle as a simple and direct way of recording both process and result.

Attention is especially called to the fact that in the pinion shaft the size is changed for each piece upon the shaft. This is



$B=1473\times 5=7365$

$T=644.8\times 21=13541$

$T_e=B+\sqrt{B^2+T^2}$

$=7365+\sqrt{7365^2+13541^2}$

$=7365+15425=22790$

$T_e=\frac{SI}{c}=\frac{Sd^3}{5.1}$ let $S=8000$

$22790=\frac{8000d^3}{5.1}$

$d^3=\frac{22790\times 5.1}{8000}=14.529$

$d=\sqrt[3]{14.529}=2.44$ say $\underline{\underline{2\tfrac{7}{16}}}$

Fig. 5.

done partly because it is desired to show the student that the shaft at each of these points should be theoretically of different size. It is also done because as a practical feature of construction it is a good plan to change the size when the fit changes, partly for reasons of production in the shop, partly for ease in slipping pieces freely endwise on the shaft until they reach their proper fit and location in the assembling of the machine.

This should not be taken as an absolute requirement in any sense. A straight shaft would be satisfactory in the present case; but the shouldered shaft is a little better construction, in a mechanical sense, and does not cost much more. Hence it is used. For the drum the straight shaft seems to answer the requirement well enough.

Bearing next to large pulley. May 21-03

see Fig. 5 and Fig. 7

$B = 1473 \times 8 = 11784$ (due to belt pull)

$B = 3458 \times 3 = 10374$ (due to load on pinion tooth)

Taking the greater value

$B = 11784$

$T = 644.8 \times 21 = 13541$

$T_e = B + \sqrt{B^2 + T^2}$

$= 11784 + \sqrt{11784^2 + 13541^2}$

$= 11784 + 17950 = 29734$

$T_e = \frac{SI}{c} = \frac{Sd_1^3}{5.1}$ let $S = 8000$

$29734 = \frac{8000}{5.1} d_1^3$

$d_1^3 = \frac{29734 \times 5.1}{8000} = 18.955$

$d_1 = \sqrt[3]{18.955} = 2.66$ say $2\frac{11}{16}$

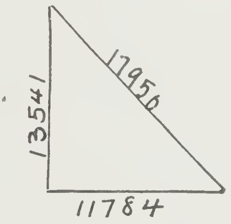


Fig. 6.

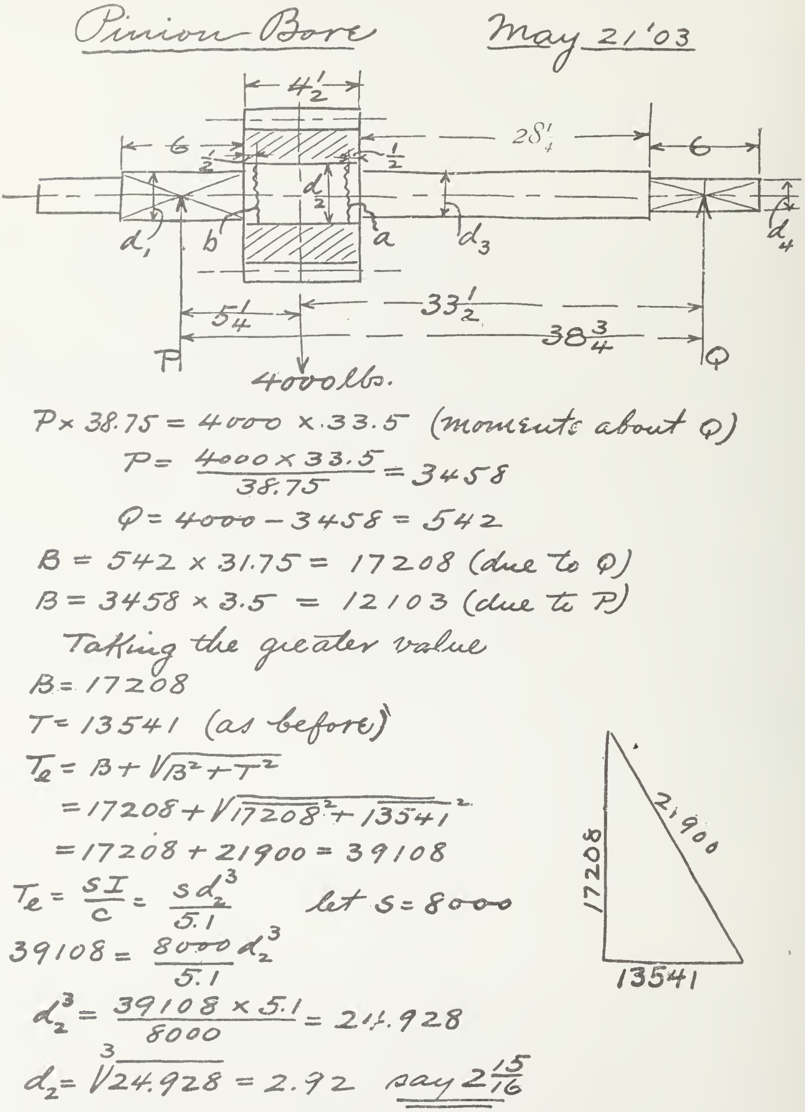
Small Pulley Bore. Fig 4.

Large Pulley Bore. Fig. 5.

Bearing Next to Large Pulley. Fig. 6.

The diameter, $2\frac{11}{16}$, as calculated, is based on the supposition that the greatest bending moment is caused by the belt pull on the overhanging pulley, that is, by the forces existing at the left-hand side of the center of the bearing.

But the pinion tooth load produces a heavy bending on the shaft in the bearing, the shaft in this case acting as a beam supported at the two bearings and having the tooth load applied as shown. If this latter effect be greater than the former, that is, if



Pinion Bore May 21 '03

$P \times 38.75 = 4000 \times 33.5$ (moments about Q)

$P = \frac{4000 \times 33.5}{38.75} = 3458$

$Q = 4000 - 3458 = 542$

$B = 542 \times 31.75 = 17208$ (due to Q)

$B = 3458 \times 3.5 = 12103$ (due to P)

Taking the greater value

$B = 17208$

$T = 13541$ (as before)

$T_e = B + \sqrt{B^2 + T^2}$

$= 17208 + \sqrt{17208^2 + 13541^2}$

$= 17208 + 21900 = 39108$

$T_e = \frac{sI}{c} = \frac{s d_2^3}{5.1}$ let $s = 8000$

$39108 = \frac{8000 d_2^3}{5.1}$

$d_2^3 = \frac{39108 \times 5.1}{8000} = 24.928$

$d_2 = \sqrt[3]{24.928} = 2.92$ say $2\frac{15}{16}$

Fig. 7.

the bending moment produced by the pinion tooth load be greater than the bending moment produced by the belt pull, then the diameter must be increased to satisfy the latter case. As is seen by the second calculation of Fig. 6, this is not the case, and the diameter stands at $2\frac{11}{16}$ as made.

Pinion Bore. Fig. 7. The pinion being a driving fit upon the shaft, reinforces the shaft to such an extent that it is hardly possible for the shaft to break off very far inside the face of the pinion; but it is quite possible that the metal of the pinion may give enough, or be a little free at the ends of the hole, so that the shaft may be broken off, say $\frac{1}{2}$ inch inside the face. In this case, it may fail from the moment of the force at the left-hand bearing or of that at the right. It may fail then at (a) or (b), depending on which section has the greater bending moment. Trying both, it is seen by the calculation that the right-hand moment is the controlling one, and it, therefore, is used.

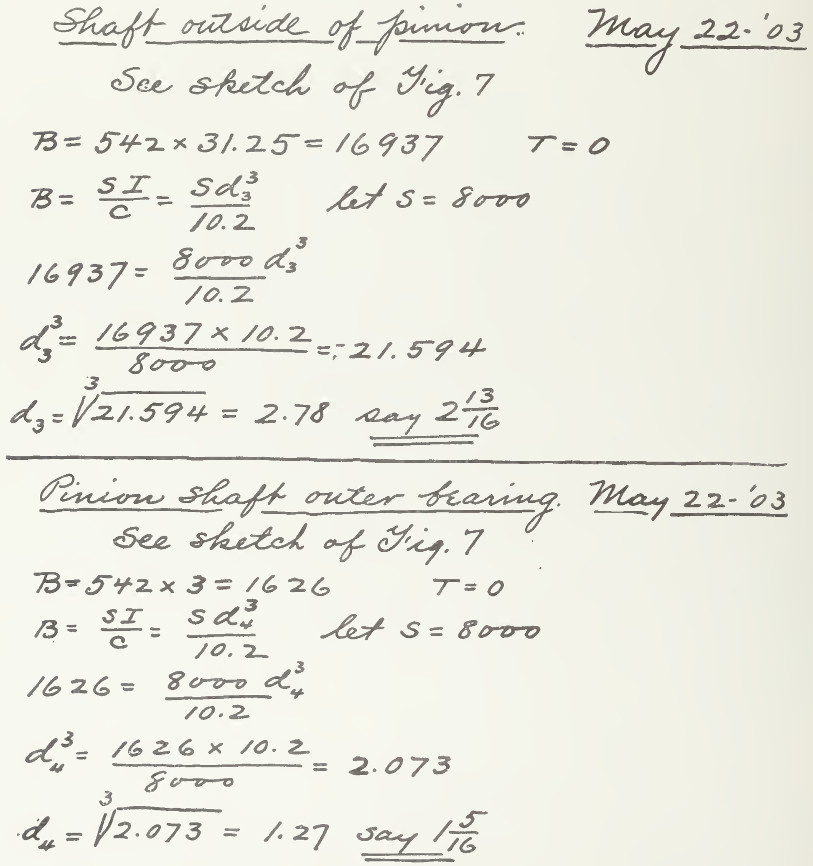
Shaft Outside of Pinion. Fig. 8. As there is no power transmitted through this portion of the shaft, there is no torsional moment in it, and the bending moment remains practically the same as inside the pinion.

The size figures about $2\frac{13}{16}$, but since there is no use in turning off material just to reduce the size to this, it is well to make it $2\frac{7}{8}$, or just smaller than the fit in the pinion.

Pinion Shaft Outer Bearing. Fig. 8. This diameter, of course, figures small, as there is no torsion in it, and the bending moment is not heavy. The practical question comes in, however, whether it is advisable to make the outer bracket different from the inner one just on account of this bearing. The commercial answer to this would probably be "No," hence the size as figured next to the pinion will be maintained ($2\frac{11}{16}$).

Drum Shaft. Fig. 9. In this case, as previously inferred, the simplest thing to do is to use a piece of straight cold-rolled steel, and make both bearings alike, the size being determined according to the worst case of loading which can occur as the rope travels from end to end of the drum. This case is evidently when the rope is at the end of its travel close to the brake, for at that time both the load on the rope and the load on the pinion tooth which is driving it are exerted upward, and produce the greatest reaction at the bearing next to the gear. The analysis of the forces for this condition is shown in Fig. 9.

Other conditions of loading would be when the brake is on and the tooth load relieved, but then the resultant of the brake strap tensions would be diagonally downward and would reduce rather than add to the rope load. Again, when the rope is at the end of the drum farthest from the gear, the load on it and the load on the pinion tooth are both exerted upward as before, but the reaction cannot be as great as in the case of Fig. 9, because the tooth load is still concentrated at the other end of the shaft and produces a relatively small reaction at the rope end.



Shaft outside of pinion. May 22-'03

See sketch of Fig. 7

$B = 542 \times 31.25 = 16937$ $\quad T = 0$

$B = \frac{SI}{c} = \frac{Sd_3^3}{10.2}$ let $S = 8000$

$16937 = \frac{8000\,d_3^3}{10.2}$

$d_3^3 = \frac{16937 \times 10.2}{8000} = 21.594$

$d_3 = \sqrt[3]{21.594} = 2.78$ say $2\frac{13}{16}$

Pinion shaft outer bearing. May 22-'03

See sketch of Fig. 7

$B = 542 \times 3 = 1626$ $\quad T = 0$

$B = \frac{SI}{c} = \frac{Sd_4^3}{10.2}$ let $S = 8000$

$1626 = \frac{8000\,d_4^3}{10.2}$

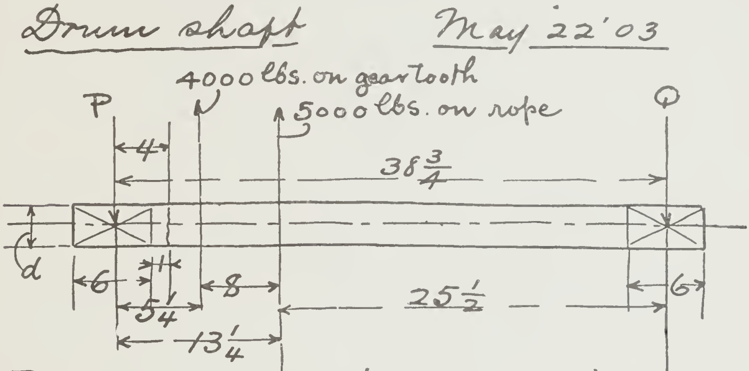
$d_4^3 = \frac{1626 \times 10.2}{8000} = 2.073$

$d_4 = \sqrt[3]{2.073} = 1.27$ say $1\frac{5}{16}$

Fig. 8.

Preliminary Layout. Fig. 10. Proceeding now with the lay out to scale, the detail of the parts may be worked out as completely as the scale of the drawing will permit. The work on this drawing may be of an unfinished, sketchy nature, but the measurements must be exact as far as they go, for this drawing is to serve as the reference sheet, from which all future detail is to be worked up.

In this layout may be worked out the sizes of the arms and hubs of pulleys and gears, the proportions of the drum and brake strap, and the general dimensions of the side brackets and the base. When the detail becomes too fine to work out to advantage on this drawing it may be worked out full size by a separate sketch, or left to be finished when it is regularly detailed. The preliminary layout, it should be remembered, is a service sheet only, a mears of carrying along the design, and not intended for



$$P \times 38.75 = 5000 \times 25\tfrac{1}{2} + 4000 \times 33\tfrac{1}{2}$$

$$P = \frac{261500}{38.75} = 6748$$

$$B = 6748 \times 4 = 26992$$

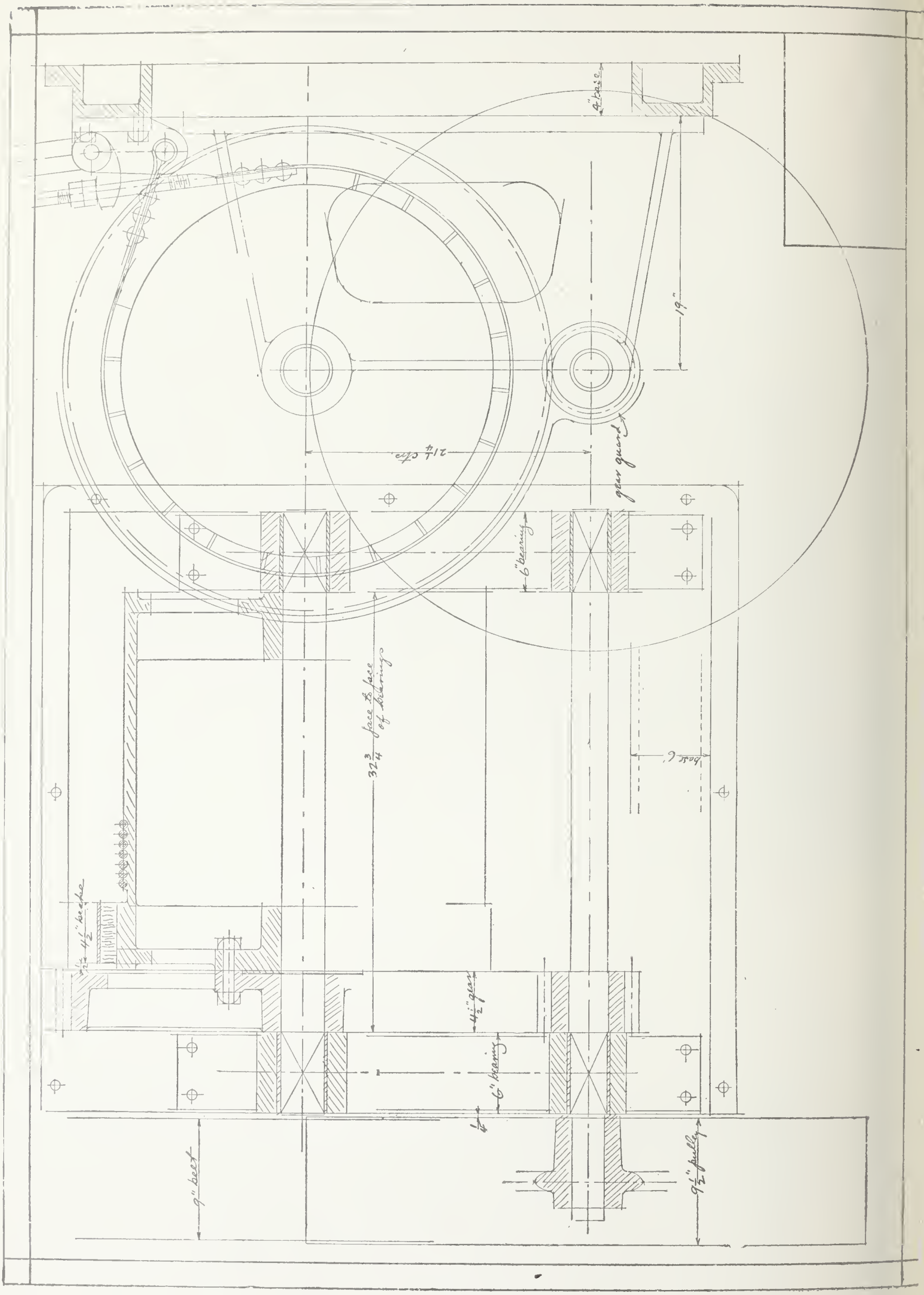
$$B = \frac{SI}{c} = \frac{Sd^3}{10.2} \qquad \text{let } S = 8000$$

$$26992 = \frac{8000\,d^3}{10.2}$$

$$d^3 = \frac{26992 \times 10.2}{8000} = 34.415$$

$$d = \sqrt[3]{34.415} = 3.25 \quad \underline{\underline{\text{say } 3\tfrac{1}{4}}}$$

Fig. 9.



4" base
19"
21¼ ctrs
gear guard
6" bearing
6" bore
32¾ face to face of bearings
4½" keepe
4½" gear
6" bearing
¼
9" belt
9½" pulley

Fig. 10.

a finished drawing. The moment that the free use of the layout is impaired by trying to make too much of a drawing of it, its value is largely lost. A designer must have some place to try out his schemes and devices, and the layout drawing is the place to do it. This drawing may be recurred to at intervals in the progress of the design, details being filled in as they are worked out, as they may control the design of adjacent parts.

As the discussion of the design of each of the members involved in the present problem can be better taken up in connection with the detail drawing of each, it will be given there, rather than in connection with the layout, although many of the proportions thus discussed could be worked out directly from the latter.

Pulleys. Fig. 11. The analysis of the forces in the belt gives, according to the calculation of Fig. 3, a tension in the tight side of 1,059 pounds, and in the slack side 414 pounds. The difference of these, or 1,059—414=645 pounds, is transmitted to the pulley and produces the torque in the shaft. Of course in the small pulley the torque is transmitted from the motor through the pulley to the belt, but both cases are the same as far as the loading of the pulleys is concerned.

The only other force theoretically acting is the centrifugal force due to the speed of the pulley. This produces tension in the rim and arms, but for the low value of 1,300 feet per minute peripheral velocity in this case may be disregarded.

Considering the arms as beams loaded at the ends, and that one-half the whole number of arms take the load, and for convenience, figuring the size of the arms at the center of the pulley, gives the following calculation for the large pulley:

$$\frac{645}{3}\times 21=\frac{S\times I}{c}=.0393\times 2{,}500\times h^3$$

Let $S=2{,}500$

" h=breadth of oval

" $.4h$=thickness of oval

$$h^3=\frac{4{,}515}{98.25}=46$$

$$h=\sqrt[3]{46}=3.6 \text{ (say 3.5)}$$

$$.4h=.4\times 3.5=1.4 \text{ (say 1 7-16)}$$

This is about all the theoretical figuring necessary on this pulley. The rim is made as thin as experience judges it capable of being cast; the arms are tapered to suit the eye, thus giving ample fastening to the rim to provide against shearing off the rim

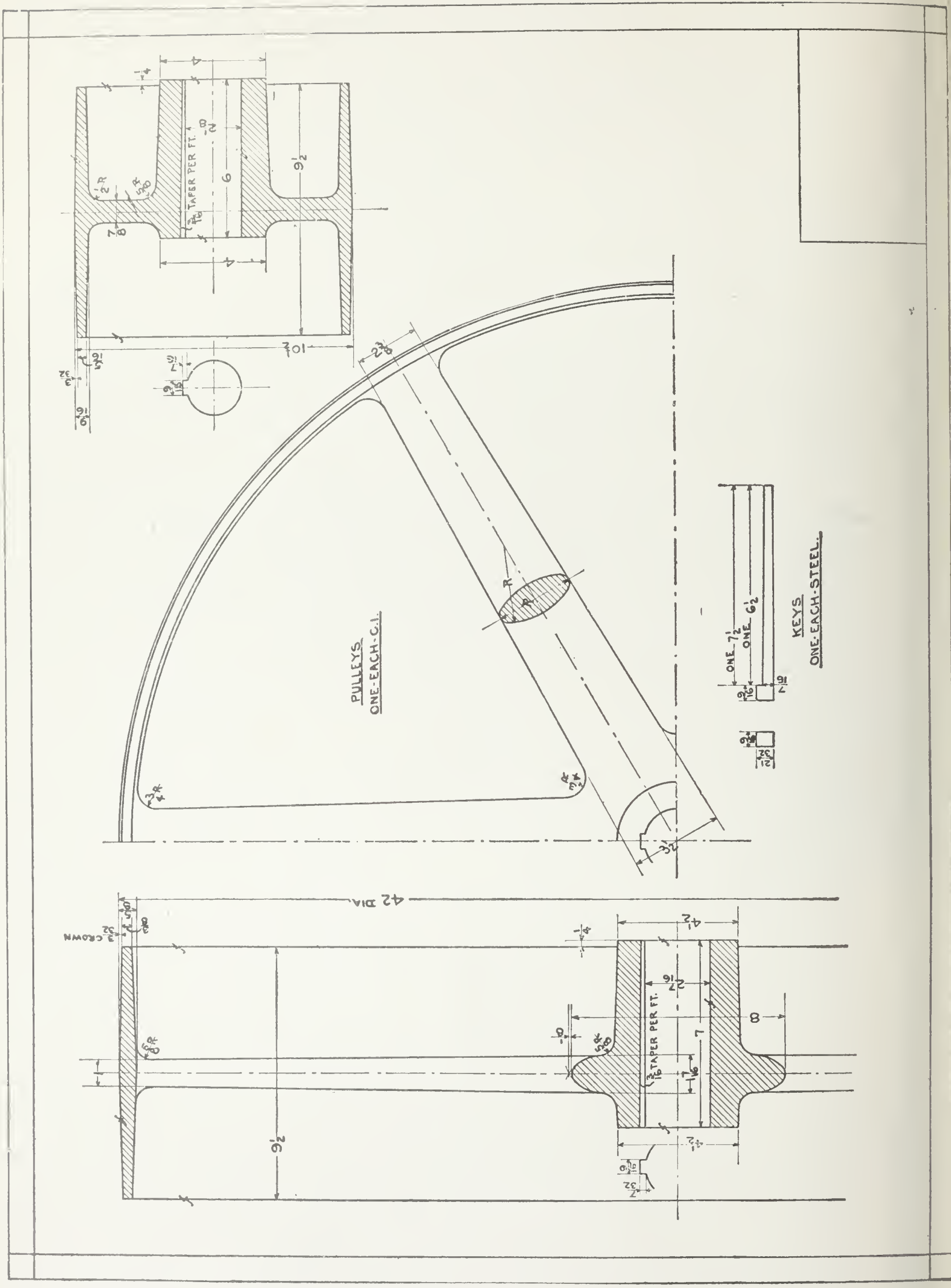

PULLEYS
ONE-EACH-C.I.
KEYS
ONE-EACH-STEEL.
ONE 7½
ONE 6½
42 DIA.
CROWN
TAPER PER FT.
TAPER PER FT.
9½
9½
10½
4
4
6
7
8
2¾
3½
R

Fig. 11.

from the arms; generous fillets join the arms to both rim and hub; and the hub is given thickness to carry the key, and length enough to prevent tendency to rock on the shaft. Uncertain strains due to unequal cooling in the foundry mold may be set up in the arms and rim, but with careful pouring of the metal they should not be serious, and the low value chosen for the fibre stress allows considerable margin for strength.

The small pulley has the same forces to withstand as the large pulley, but on account of its small diameter there is not room enough for arms between the rim and the hub, hence it is made with a web. The web cannot be given any bending by the belt pull, the only tendency which exists in this case being a shearing where the web joins the hub. This shearing also exists throughout the web as well, but at other points farther from the center it is of less magnitude, and moreover, there is more area of metal to take it. The natural way to proportion the thickness of the web is to give it an intermediate thickness between that of the hub and rim, thus securing uniform cooling, and then figure the stress as a check. Making this value $\frac{7}{8}$ inch gives a shearing area of $\frac{7}{8}$ multiplied by the circumference of the hub, which is $3.1416 \times 4 = 12.56$. The shearing force at the hub is $\frac{645 \times 5.25}{2} = 1{,}693$ pounds. Equating the external force to the internal resistance

$$1{,}693 = \tfrac{7}{8} \times 12.56 \times S$$

$$S = \frac{1{,}693 \times 8}{7 \times 12.56} = 154 \text{ pounds per square inch (approx.)}$$

This is a very low figure, even for cast iron, hence the web is amply strong. The rim and hub are proportioned as for the large pulley.

The keys are taken from the standard list. They may be checked for shear, crushing in the hub, and crushing in the shaft, but the hubs are so long that it is at once evident without figuring that the stress would run very low in both cases.

Gears. Fig. 12. The analysis of the forces acting on the gears has been given on page 28, 4,000 pounds being taken at the pitch line. Using this same value, and choosing a T-shaped arm as a good form for a heavily loaded gear like the present one, let us consider that the rim is stiff enough to distribute the load

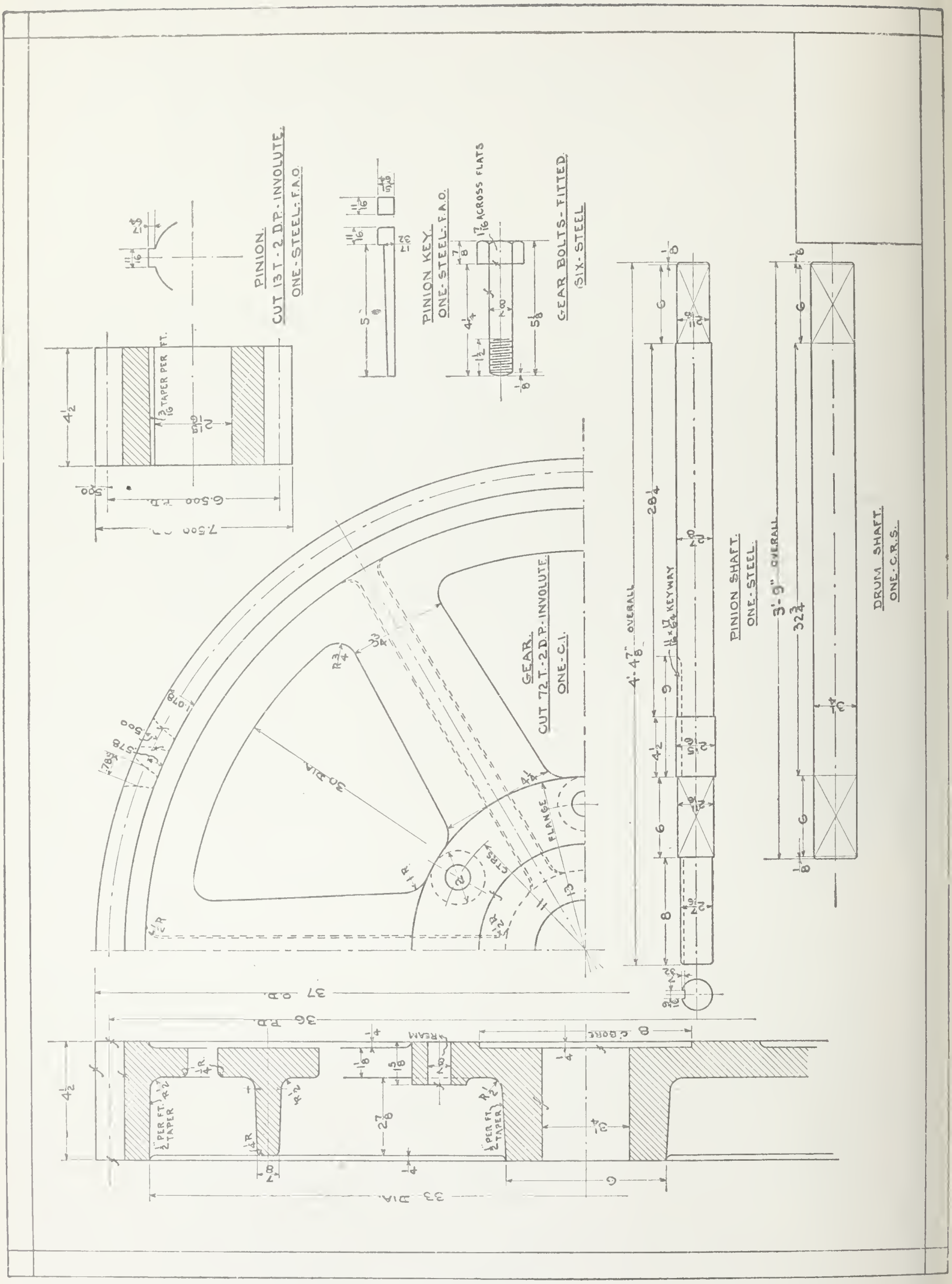

PINION.
CUT 13 T. 2 D.P. INVOLUTE.
ONE-STEEL: F.A.O.
3/16 TAPER PER FT.
6·500 P.D.
7·500 O.D.
PINION KEY.
ONE-STEEL: F.A.O.
17/16 ACROSS FLATS
GEAR BOLTS-FITTED.
SIX-STEEL
GEAR.
CUT 72 T. 2 D.P. INVOLUTE.
ONE-C.I.
30 DIA
FLANGE
CTRS
37 O.D.
36 P.D.
33 DIA
6 BORE
REAM
1/2" PER FT. TAPER
4' 4 7/8" OVERALL
11/16 × 17/64 KEYWAY
PINION SHAFT.
ONE-STEEL.
3' 9" OVERALL
32 3/4
DRUM SHAFT.
ONE-C.R.S.

Fig. 12.

equally between all the arms, and that each acts as a beam loaded at the end with its proportion of the tooth load. Before we can determine the length of these arms, however, we must fix upon the size of the flange which is to carry the driving bolts. This is taken at 13 inches. It could be smaller if desired, but drawing the bolts in toward the center increases the load on them, and 13 inches seems reasonable until it is proved otherwise. This makes the maximum moment which can come on an arm $\frac{4{,}000\times 11.5}{6}=7{,}666$ inch-pounds.

Now it is evident that the base of the T arm section, which lies in the plane of rotation, is most effective for driving, and that the center leg of the T does not add much to the driving capacity of the arm, although it increases the lateral stiffness of the arm, as well as providing in casting a free flow of metal between the rim and the hub. Hence the simplest way of treating the section of the arm for strength is to consider the base of the T only, of rectangular section, breadth b, and depth h, for which the internal moment of resistance is $\frac{S\times b\times h^2}{6}$.

Also, it is simplest to assume one dimension, say the breadth, and the allowable fibre stress, and figure for the depth. Taking the breadth at $1\frac{1}{8}$ inches, which looks about right, and the fibre stress at 2,500, and equating the external moment to the internal, we have

$$7{,}666=\frac{2{,}500\times 1.125\times h^2}{6}$$

$$h^2=\frac{6\times 7{,}666}{2{,}500\times 1.125}=16.4$$

$$h=\sqrt{16.4}=4.05\text{ (say }4\tfrac{1}{4})$$

Drawing in this size, and tapering the arm to the rim as in the case of the pulleys, making the depth of the rim according to the suggested proportions given in Part II, "Gears," giving the center leg of the T a thickness of $\frac{7}{8}$ inch tapering to 1 inch, and heavily filleting the arms to the rim and center flange, we have a fairly well proportioned gear.

The next thing to determine is the size of the driving bolts. The circle upon which their centers lie may be 11 inches in diam-

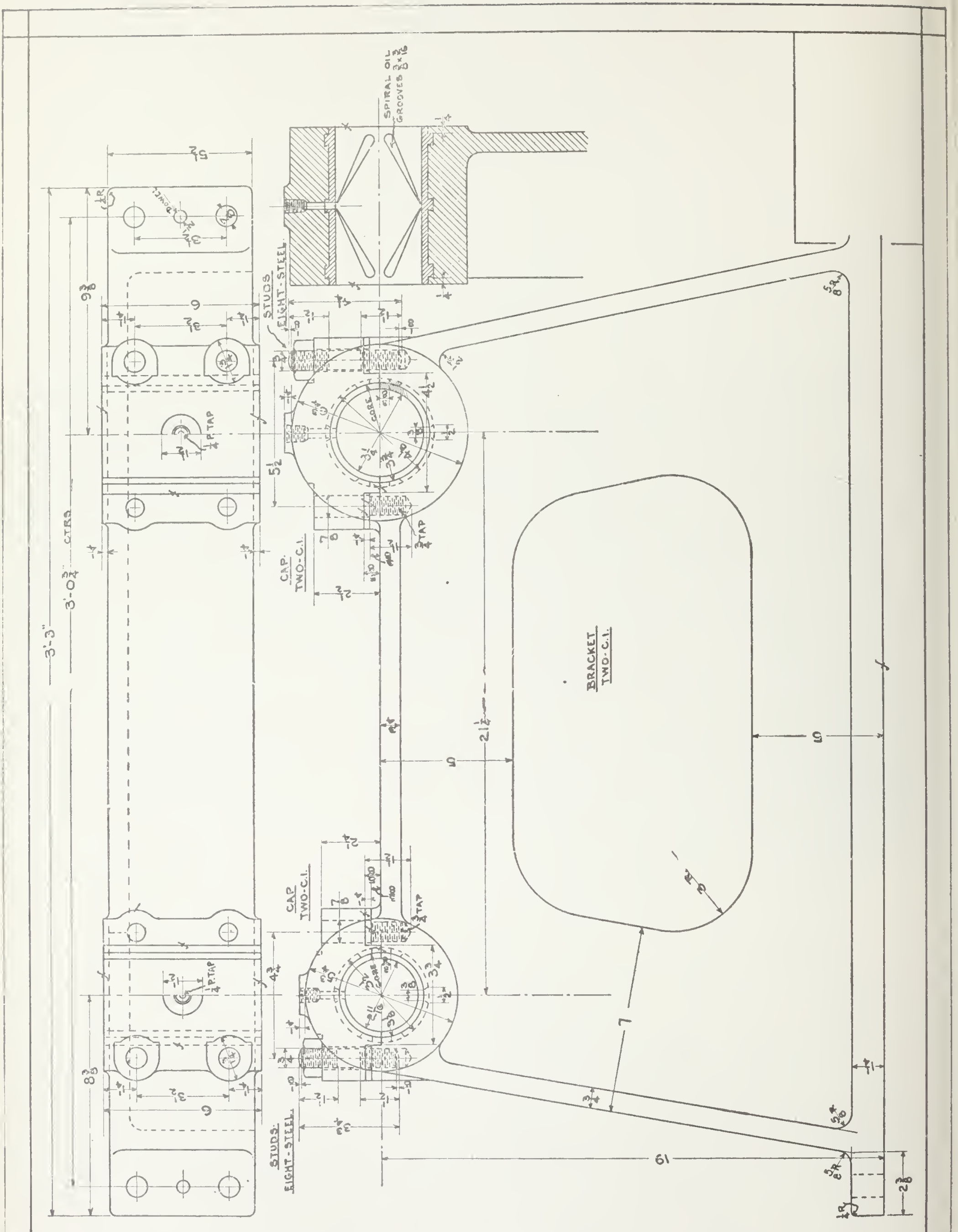

BRACKET
TWO-C.I.
CAP
TWO-C.I.
STUDS
EIGHT-STEEL
SPIRAL OIL GROOVES
CORE
¼ P. TAP
¾ TAP
CTRS
3'-3"
3'-0¾"
21¼
9⅜
8⅜
5½
19
7
5

Fig. 13.

eter, and there will naturally be six bolts, one between each arm. These bolts are in pure shear, and the material of which they are to be made ought to be good for at least 8,000 pounds per square inch fibre stress. The force acting at the circumference of an 11-inch circle would be $\frac{4,000\times 18}{5.5}=13,091$ pounds.

Equating the load on each bolt to the resisting shear gives

$$\frac{13,091}{6}=8,000\times A=\frac{8,000\times 3.1416\times d^2}{4}$$

Let A=area resisting shear.
Let d=dia. of bolt.
Then $A=\frac{\pi\, d^2}{4}$

$$d^2=\frac{4\times 13,091}{6\times 8,000\times 3.1416}=.35$$

$d=\sqrt{.35}$ (say .6) ⅝-inch bolts would do.

But ⅝-inch bolts are pretty small to use in connection with such heavy machinery. They look out of proportion to the adjacent parts. Hence ⅞-inch bolts have been substituted as being better suited to the place in spite of the fact that theoretically they are larger than necessary. The extra cost is a small matter. These bolts may crush in the flange as well as shear off, but as there is an area of $\frac{7}{8}\times 1\frac{5}{8}=1.422$ square inches to take $\frac{13,091}{6}=2,182$ pounds, the pressure per square inch of projected area is only $\frac{2,182}{1.422}=1,534$ pounds, which is very low.

This gear needs no key to the shaft because all the power comes down the arms and passes off to the drum through the bolts, thus putting no torsional stress in the shaft. The face of the flange is counterbored so as to center the gear upon the drum, without relying upon the fit of the gear upon the shaft to do this.

The pinion is solid and needs no discussion for its design.

Brackets and Caps. Fig. 13. As the size of the drum shaft was determined by considering the rope wound close up to the brake, thus giving in combination with the load on the gear tooth the maximum reaction at the bearing as 6,748 pounds, the cap and bolts should be designed to carry the same load.

For a bearing but 6 inches long, two bolts are sufficient under ordinary conditions and might perhaps do for this case. The load is pretty heavy, however, and it is deemed wise to provide four bolts, thus securing extra rigidity, and permitting the use of bolts of comparatively small size. If the load were distributed equally over all the bolts each would take one-fourth of the whole load, but it is not usually safe to figure them on this basis, because it is difficult to guarantee that each bolt will receive its exact share of stress. Assuming that the two bolts on one side take $\frac{2}{3}$ the whole load instead of $\frac{1}{2}$, which provides for this uncertain extra stress, each bolt must take care of $\frac{1}{3}$ of 6,748, or 2,249, pounds. Allowing 8,000 pounds per square inch fibre stress calls for an area at the root of the thread of $\frac{2,249}{8,000} = .281$ square inch. Consulting a table of bolts we find that the next standard size of bolt greater than this is $\frac{3}{4}$, which gives an area of .302 square inch.

Choosing this size as satisfactory, the bolts should be located as close to the shaft as will permit the hole to be drilled and tapped without breaking out. A center distance of $5\frac{1}{2}$ inches accomplishes this result. The distance between centers in the other direction is somewhat arbitrary, although the theoretical distance between the bolt and the end of the bearing to give equal bending moment at the center of the cap and at the line of the bolts is about $\frac{5}{24}$ of the length, or $\frac{5}{24}$ of $6 = 1\frac{1}{4}$ inches. This proportion answers well for the present case, although for long caps it brings the bolts too far in to look well.

The thickness of the cap may be determined by assuming it to be a beam supported at the bolts and loaded at the middle. This is not strictly true, for the load is distributed over at least a portion of the shaft diameter; moreover, the bolts to some extent make the beam fixed at the ends. It being impossible to determine the exact nature of the loading, we may take it as stated, supported at the ends and loaded in the middle, and allow a higher fibre stress than usual, say 3,500. The longitudinal section at the middle of the cap is rectangular, of breadth 6 inches, and depth unknown, say h. The equation of moments is

$$\frac{W \times l}{4} = \frac{S \times I}{c} = \frac{S \times b \times h^2}{6}$$

$$\frac{6,748 \times 5.5}{4} = \frac{3,500 \times 6 \times h^2}{6}$$

$$h^2 = \frac{6 \times 6,748 \times 5.5}{4 \times 3,500 \times 6} = 2.65$$

$$h = \sqrt{2.65} = 1.62 \text{ } (1\tfrac{1}{2} \text{ will probably answer})$$

For the other bearing next to the pinion, the load on the tooth acts downward, and the resultant pull of the belt is nearly horizontal, hence the cap and bolts must stand but little load, and calculation would give minute values. In a case like this it is well to make the size the same as for the larger bearing, unless the construction becomes very clumsy thereby. This saves changing drills and taps in making the holes, and preserves the symmetry of the bracket. The $\frac{3}{4}$-inch bolts are good proportion for the smaller bearing, hence that size will be maintained throughout.

The body of the bracket is conveniently made with the web at the side and horizontal ribs extending to the outside. The load due to the rope is carried directly down the side ribs and web into the bottom flanges and to the bolts. The analysis of the forces on these bolts is shown in Fig. 14. It is evident from the figure that the resultant belt pull tends to hold the bracket down, while the load on the rope tends to pull it up, the point about which it tends to rotate being the corner furthest from the drum. It is also evident that the bolts nearest this corner can have little effect on the holding down, because their leverage is so small about the corner, hence we shall assume that the pair of bolts at the right-hand end of the bracket takes all the load. The belt pull, being horizontal, tends to slide the bracket along the base, but this tendency is small, and at any rate is easily taken care of by the two dowel pins, which are thus put in shear.

The load on the bolts being 4,954 pounds, a heavy bending moment is thrown on the flange of the bracket, tending to break it off at the root of the fillet. The distance to the root of the fillet is $\frac{3}{4}$ inch; the section tending to break is rectangular, of breadth $5\frac{1}{2}$ inches, and unknown depth h. The equation of moments is

$$W \times l = \frac{S \times I}{c} = \frac{S \times b \times h^2}{6}$$

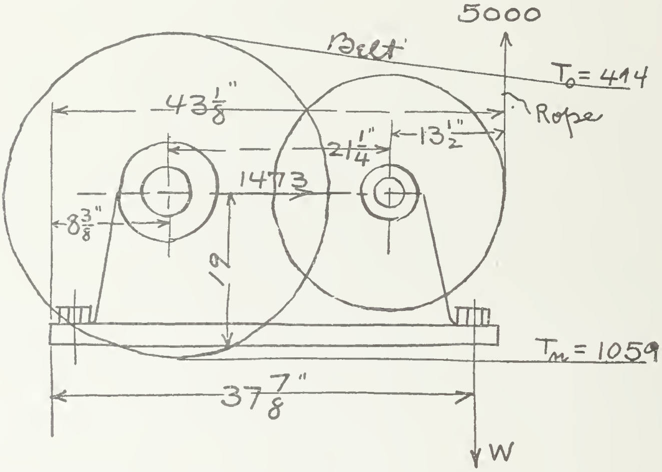
$$\frac{4{,}954 \times 3}{4} = \frac{2{,}500 \times 5.5 \times h^2}{6}$$

$$h^2 = \frac{6 \times 4{,}954 \times 3}{4 \times 2{,}500 \times 5.5} = 1.62$$

$$h = \sqrt{1.62} = 1.3 \text{ (say } 1\tfrac{1}{4}\text{)}.$$

The thickness of the web and ribs of this bracket is hardly capable of calculation. The figure $\frac{3}{4}$ inch has been chosen in proportion to the size of the large drum bearing, giving ample stiffness and rigidity, and permitting uniform flow and cooling of the metal in the mold. The opening in the center is made merely to save material, as in that part little stress would exist, the two sides

Bracket base bolts May 25-1903



$$5000 \times 43\tfrac{1}{8} = 37\tfrac{7}{8} \times W + 1473 \times 19$$

$$W = \frac{5000 \times 43.125 - 1473 \times 19}{37.875}$$

W = 4954 lbs on two bolts

$\frac{4954}{2}$ = 2477 lbs per bolt.

$\frac{2477}{8000}$ = .31 sq. in. area at 8000 fib. stress.

Area at root of thread for a $\frac{3}{4}$ bolt is .28

Hence $\frac{3}{4}$ bolts will answer.

Fig. 14.

carrying the load down to the base bolts, and the top serving as a tie between the bearings.

This bracket might be made with the web in the center of the bearings instead of at the side, in which case the expense of the pattern would be slightly greater. It could also be made of closed box form, but would in that case probably weigh more than as shown.

Drum and Brake. Fig. 15. The analysis of the forces acting on the drum is simple, but its theoretical design is more complicated. It is evident that the drum acts as a beam of hollow circular cross section, and that its worst case of loading is when the rope is at or near the middle of the drum length. At the same time the metal of this circular cross section is in a state of torsion between the free end of the rope and the driving gear, due to the load on the gear tooth and the reaction of the rope. Also the wrapping of the rope around the drum tends to crush the metal of the section beneath it, the maximum effect of this action being near the free end of the rope where its tension has not been reduced by friction on the drum surface.

Now the "mechanics" to solve the problem of these three combined actions is rather complicated. It can be at least approximately solved, however, for it satisfies fairly well the case of combined compression and shear. But on a further study of this particular case, it is seen at once that the diameter of the drum is relatively large with respect to its length, which means that the thickness of the metal may be very small and yet give a large resisting area, or value of "I," both in direct bending as well as torsion; also it is so short that the external bending moment will be small. The practical condition now comes in, that the drum can be safely cast only when the thickness of the metal is at a minimum limit, for the core may be out of round, not set centrally, or by some other variation produce thin spots or even develop holes reaching out into the rope groove, discovered only when the latter is turned in the lathe.

Hence it seems reasonable and safe in this case to make the thickness of the drum depend simply upon the crushing caused by the wrapping of the rope around it, and we shall take the coil nearest the free end of the rope, assuming that it carries the full load of 5.000 pounds throughout one complete wrap around the drum.

The area resisting the crushing action may be considered to be that of the cross section of a ring, of width equal to the pitch

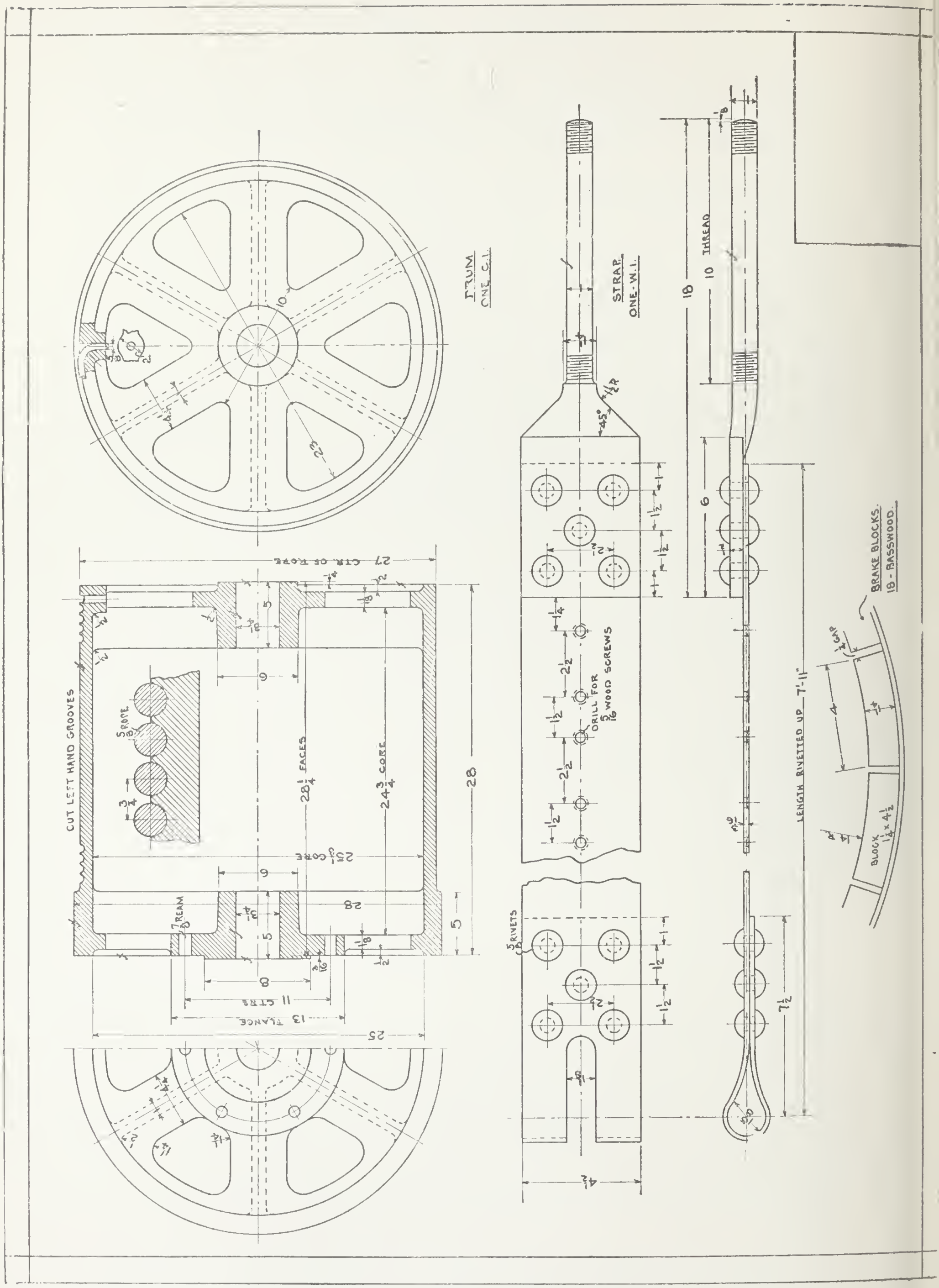

DRUM
ONE C.I.
STRAP
ONE-W.I.
BRAKE BLOCKS.
18 - BASSWOOD.
CUT LEFT HAND GROOVES
5/8 ROPE
27 CTR. OF ROPE
28 1/4 FACES
24 3/4 CORE
25 1/4 CORE
28
13 FLANGE
11 CTRS
25
7/8 REAM
DRILL FOR 5/16 WOOD SCREWS
5/8 RIVETS
10 THREAD
18
LENGTH RIVETTED UP 7'-11"
BLOCK 1 1/4 x 4 1/2
1/4 GAP
45°
7 1/2

Fig. 15.

of the groove. Assuming that $\frac{5}{8}$ inch is the least thickness which can be safely allowed under the groove for casting purposes, let us figure the crushing fibre stress to see if this is sufficiently strong. Disregarding the small amount of metal existing above the bottom of the groove, this gives the area to resist the crushing $\frac{5}{8}\times\frac{3}{4}=\frac{15}{32}$, or .47 inch. Since there are two of these sections and the rope acts on both sides, the equation of forces is:

$$5{,}000\times 2 = S\times .47\times 2$$

$$S=\frac{5{,}000\times 2}{.47\times 2}=10620 \text{ pounds per square inch.}$$

This, for cast iron, in pure crushing, allows plenty of margin for the extra bending and torsional stress, which for such a considerable thickness would be slight.

The above case indicates a method of reasoning much used in designing machinery, which while following out the specified routine of thought as previously given in these pages, stops short of elaborate and minute theoretical calculation when such is obviously unnecessary. If a drum of great length were to be designed, and of small diameter, the same method of reasoning would deduce the fact that the design should be based on the bending and the torsional moments, the thickness in such a case being so great to withstand these that the intensity of the crushing due to wrap of the rope becomes of inappreciable value.

The remaining points of design of the drum are determined from practical considerations and judgment of appearance. The ribs behind the arms are put in to give lateral stiffness and guard against endwise collapse. The arms are subject to the same bending as those of the gear, but as they are equally heavy it is not necessary to calculate them. The flange at the driving end is of course matched to that already designed for the gear. The rope is intended to be brought through the right-hand end with an easy bend and the standard form of button wedged on to prevent its pulling through.

This drum would probably be cast with its axis vertical, and the driving flange down to secure sound metal at that point. Heavy risers would be left at the other end to secure soundness where the rope is fastened. Drums are often cast with the axis horizontal, but the vertical method is more certain to produce a sound casting. The grooves should be turned from the solid metal, partly because it is a difficult matter to cast them, but principally because the rope should run on as smooth surface as possible to avoid undue wear. On drums which carry chain instead of wire rope the grooves are sometimes cast with success, although even in this case the turned groove is generally preferable.

The brake consists of a wrought-iron band to which are fastened wooden blocks, the iron band giving the requisite strength while the blocks give frictional grip on the drum surface and can be easily replaced when worn. As in the designing of a belt the object in view is the grip on the pulley surface by the leather to enable power to be transmitted from the belt to the pulley, so in the case of the brake if we put the proper tension in the strap it can be made to grip the brake drum so tightly that motion between it and the drum cannot occur. The latter case is really the reverse of the first, if the driven pulley be considered, but is identical with the case of the driving pulley, in which the power is transmitted from the pulley to the belt. Of course in the case of the brake no power is transmitted, as when the brake holds no motion occurs, but the principle of the relative tensions in the strap is the same as for the belt.

Since the brake drum surface is 28 inches in diameter, the load at that surface which the brake must hold is

$$P = \frac{5{,}000 \times 27}{14 \times 2} = 4{,}821 \text{ pounds.}$$

We have then the following calculation corresponding exactly to that of the belt given in Fig. 3.

$\log. \frac{T_n}{T_o} = 2.729 \times \mu \times n$ Let $\mu = .25$

“ $n = .75$

$T_n - T_o = P = 4{,}821$

$\log. \frac{T_n}{T_o} = 2.729 \times .25 \times .75 = 0.512$ (for which the natural number is 3.25).

Then $\frac{T_n}{T_o} = 3.25$ $T_o = \frac{T_n}{3.25}$

$T_n - T_o = 4{,}821$ $T_n - \frac{T_n}{3.25} = \frac{2.25 \times T_n}{3.25} = 4{,}821$

$T_n = \frac{4{,}821 \times 3.25}{2.25} = 6{,}963$ pounds (say 7,000)

$T_o = 6{,}963 - 4{,}821 = 2{,}142$ pounds (say 2,200)

The tight end of the strap must then be capable of carrying a load of 7,000 pounds, and since the width has already been taken at $4\frac{1}{2}$ inches, the problem is to find the necessary thickness. Equating the external load to the internal resistance we have

$7{,}000 = A \times S$ Let t = thickness

" S = fibre stress = 12,000

$$7{,}000 = 4.5 \times t \times 12{,}000$$

$$t = \frac{7{,}000}{4.5 \times 12.000} = .13 \text{ inch}$$

This, however, can be but a preliminary figure, for the riveting of the strap will take out some of the effective area, and the thickness will have to be increased to allow for this. Suppose on the basis of this figure we assume the thickness at a slightly increased value, say $\frac{3}{16}$ inch, and proceed to calculate the rivets.

A group of five rivets will work in well for this case, which gives $\frac{7{,}000}{5} = 1{,}400$ pounds per rivet. A safe shearing fibre stress is 6,000, hence the area necessary per rivet is $\frac{1{,}400}{6{,}000} = .23$ square inch. This comes nearest to the area $\frac{9}{16}$ diameter, but for the sake of using the more general size of rivet ($\frac{5}{8}$ inch) the latter is chosen, for which the area is .30.

We must now try these rivets in a $\frac{3}{16}$-inch plate for their safe bearing value. The projected area of a $\frac{5}{8}$-inch hole in a $\frac{3}{16}$-inch plate is $\frac{5}{8} \times \frac{3}{16} = .117$ square inch. $\frac{1{,}400}{.117} = 11{,}965$ (15,000 would be safe)

Taking out two $\frac{5}{8}$-inch rivets from the full width of $4\frac{1}{2}$ inches leaves $4\frac{1}{2} - (2 \times \frac{5}{8}) = 3.25$, and makes the net area of strap to take stress $3.25 \times \frac{3}{16} = .61$ square inches. Re-calculating the fibre stress for this area gives

$$7{,}000 = .61 \times S$$

$$S = \frac{7{,}000}{.61} = 11{,}475$$ (which approximates the previous value of 12,000).

The slack end of the strap has to take but 2,200 pounds, hence a different calculation might be made for this end giving smaller rivets; but as it is impractical to change the thickness of the strap to meet this reduced load, it is well to maintain the same proportion of joint as at the tight end. The spacing of the rivets in both

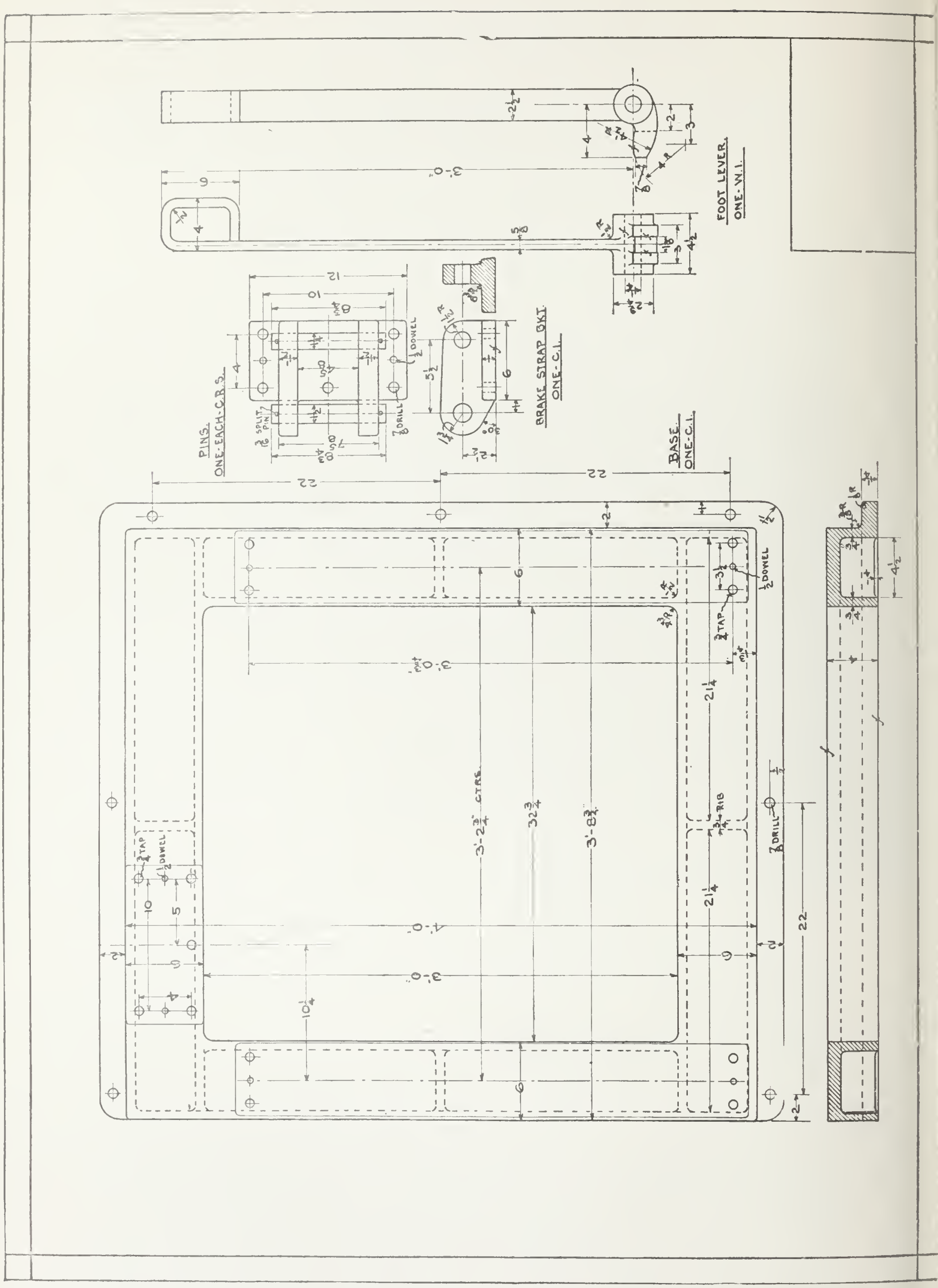

FOOT LEVER.
ONE-W.I.
BRAKE STRAP BKT.
ONE-C.I.
PINS.
ONE-EACH-C.R.S.
BASE.
ONE-C.I.
3'-0"
3'-2¾" CTRS
32¾
3'-8¾"
4'-0"
22
21¼
10¼
¾ RIB
⅞ DRILL
¾ TAP
½ DOWEL
3/16 SPLIT PIN
⅞ DRILL

Fig. 16.

cases follows the ordinary rule allowing at least three times the diameter of the rivet as center distance, and one-half this value to the edge of the plate.

The threaded end of the forging on the strap also has to carry the load of 2,200 pounds, for which a size smaller than 1 inch would suffice. It is natural, however, for the sake of general proportion to make the bolt as strong as the strap, and a 1-inch bolt gives an area of .52 square inch, nearly equalling the value of .61 net area of strap noted above.

Base, Brake=Strap Bracket and Foot Lever. Fig 16. The base cannot be definitely calculated, and can best be proportioned by judgment. It must not distort, twist, or spring in any way to throw the shafts out of line. The area in contact with the foundation upon which it rests must be ample to carry the weight of the whole machine with a low unit pressure. Although the form shown is perfectly practicable to cast and machine, and is simple and rigid, yet it is questionable if a bolted-up construction, say of four pieces, might not be equally rigid and yet involve greater facility of production in both the foundry and machine shop on account of the reduced sizes of parts to be handled. This is a question which depends on the equipment and methods of the individual shop, and is an illustration of the practical control of design by manufacturing conditions.

The brake-strap bracket and foot lever, also shown in this figure, are examples of machine parts which are quite definitely loaded, and the designing of which is a simple matter. Further discussion of their design is not made, the student being given opportunity for some original thought in determining the forces and moments that control their design.

Gear Guard and Brake=Relief Spring. In exposed machinery of this character it is desirable to cover over the gears with a guard to prevent anything accidentally dropping between the teeth and perhaps wrecking the whole machine. This guard is not shown, as it involves little of an engineering nature to interest the student. It could readily be made of sheet metal or light boiler plate, bent to follow the contour of the gears and fastened to the top flange of the main bracket.

If the brake be not automatically supported at its top it will lie with considerable pressure, due to its own weight, on the brake surface when it is supposed to be free from it, and by the friction thereby created will produce a heavy drag and waste of power. A spring connection fastened to an overhead beam is a simple way of accomplishing the desired result. A flat supporting strap carried out from the gear guard, having some degree of spring in it, is a neater method of solving the problem. The spring should be just strong enough to counterbalance the weight of the strap and yet not resist to an appreciable degree the force applied to throw the brake on.

GENERAL DRAWING.

The last step in the process of design of a machine is the making of the assembled or general drawing. This should be built up piece by piece from the detail drawings, thereby serving as a last check on the parts going together. This drawing may be a cross section or an outside view. In any case it is not wise to try to show too much of the inside construction by dotted lines, for if this be attempted, the drawing soon loses its character of clearness, and becomes practically useless. A general drawing should clearly **hint at, but not specify,** detailed design. It is just as valuable a part of the design as the detail drawing, but it cannot be made to answer for both with any degree of success. A good general drawing has plenty of views, and an abundance of cross sections, but few dotted lines.

The general drawing of the machine under consideration is left for the student to work up from the complete details shown. It would look something like the preliminary layout of Fig. 10, if the same were carefully carried out to finished form. A plain outside view would probably be more satisfactory in this case than a cross section, as the latter would show little more of value than the former. The functions which the general drawing may serve are many and varied. Its principal usefulness is, perhaps, in showing to the workman how the various parts go together, enabling him to sort out readily the finished detail parts and assemble them, finally producing the complete structure. Otherwise the making of a machine, even with the parts all at hand, would be like the putting together of the many parts of an intricate puzzle, and much time would be wasted in trying to make the several parts fit, with perhaps never complete success in giving each its absolutely correct location.

The general drawing also gives valuable information as to the total space occupied by the completed machine, enabling its location in a crowded manufacturing plant to be planned for, its connection to the main driving element arranged, and its convenience of operation studied.

In some classes of work it is a convenient practice to letter each part on the general drawing, and to note the same letters on the specification or order sheet, thus enabling the whole machine to be ordered from the general drawings. This is a very excellent service performed by the general drawing in certain lines of work, but for such a purpose the drawing is quite inapplicable in others.

Merely as a basis for judgment of design, the general drawing fulfils an important function in any class of work, for it approaches the nearest possible to the actual appearance that the machine will have when finished. A good general drawing is, for critical purposes, of as much value to the expert eye of the mechanical engineer as the elaborate and colored sketch of the architect is to the house builder or landscape designer.

From the above it is readily understood that the general drawing, although a mere putting together of parts in illustration, is yet of great assistance in producing finished and exact machine design.

GENERAL COMMENTS ON PRECEDING PROBLEM.

After following through the detail of work as given in the preceding pages, it is worth while to stop for a moment and take a brief survey or review of the subject as illustrated therein.

If the text be carefully studied it will be seen that in every part to be designed the same routine method has been followed, regardless of the final outcome. In some cases it may seem a roundabout procedure to follow a train of thought that finally ends in a design apparently based on purely practical judgment, the theory having had but very little if any influence. The question at once arises—Why not use the empirical rule or formula in the first place? Why not make a good guess at once? Why not save all the time and energy devoted to a careful analysis and theory, if we are finally to throw them away and not base our design on them?

The principle to be noted in this connection is, that it is just as fatal to good design to rely upon bare experience and upon judgment alone, as it is to construct solely according to what pure theory tells us. There are many things in the operation of machinery that are totally inexplicable from the purely practical point of view, and will forever remain so until we analyze them and theorize on them. Many good things in machinery have been the result of what might be called "reversed" machine design. When a new machine is started, it frequently, or we might almost say always, fails to do its work just as it is expected to do it. This is because some little point of design is bad, owing to the inability of drawings, however good they may be, to show all that the machine itself in bodily form and in motion shows.

Now, if our analysis and theory have been good in the designing process, it is almost sure that we can very readily analyze and theorize on the trouble that exists when the machine is finished, can detect the weakness, and can correct it with comparatively small change in the general design. This is "reversed" machine design.

If, on the contrary, we have based our design purely on guesswork, allowing our fancy full and free play to work out the details without further basis, we may consider ourselves lucky if the machine runs at all. This, however, is not the worst of the situation. If the machine does actually operate, even as well as it might reasonably be expected to, but still has the usual difficulty of some little kink or hitch that was not expected, then, as a result of the method upon which the whole thing has been constructed, we have no definite plan of action to proceed upon. We must try first this, then that scheme to obviate the trouble. We may be fortunate enough to "strike it" the first time; we may never strike it. It is doubtful if the machine ever can be made to work at highest efficiency; and if fairly good results be finally obtained we never know the reason why, and have nothing on which to base any future action or design.

This haphazard process is not machine design at all, either in name or in result.

As has previously been stated in these pages, there is no such thing as too much analysis or theory in the designing of machinery. Even if we carefully analyze, theorize with rigorous exactness, and then practically modify our construction to such a point that the original theoretical shape is almost or entirely lost, the apparently roundabout process is not in vain, for we are in perfect control of our design. We know exactly what it has to take in the way of forces, blows and vibrations. We know what its ideal shape should be. We know where we can practically modify its form without weakening it excessively or adding excess of material. In other words we know all about it, and therefore know exactly what we can do with it; and whether it follows in its shape the outline that pure theory gives it or some other outline, it is nevertheless well designed.

"Reversed" machine design, as described above, based on observation and experiment with regard to machines already in operation, is just as impossible without exact analysis and theory as is original design based merely on mechanical ideas in the abstract. The method once learned and made a habit of mind will produce results with equal facility in either case, and results are what the mechanical world is seeking.

Another point worth noting in the progress of the problem as given is the absolute necessity of possessing some knowlege of Mechanics. The more of this subject the designer can have at his finger ends, the more ready and successful will he be in all problems of Machine Design. However, the principles of forces and moments clearly understood, and the application of the same in the all-important subject, "Strength of Beams," constitute a fund of information that will give a splendid start and a good working basis for simple designs. It should always be remembered that a complicated design is little more than a combination of simple designs, and if one has the ability to dissect and analyze what seems at first like a bewildering maze of parts, complication is speedily changed to simplicity.

Common sense goes a long way in good designing. There is nothing mysterious about the process If the beginner will only avoid doing things that are foolish and ridiculous on their very face, if he will exercise the same judgment that he uses in the daily affairs of his life and will mix in something of mechanics and mechanical method, he will be on the direct road to success in the art.

Good drawing is an essential element of good design, and it is especially urged that the sketches and drawings as reproduced in the preceding text be studied with this in mind. By a good drawing is meant not a showy piece of work, finely shaded or artistically lettered, but an exact layout, definite and measurable, correctly dimensioned if in detail, and meaning exactly what it says. Machine design is an exact science, and the designer cannot shirk responsibility by permitting his work to be shiftless and loose. If he cannot delineate clearly and in definite form what he determines in his mind the structure should be, then it is purely good luck if he achieves success, and it may safely be asserted that the success is due to some subsequent care and finished design added to his feeble effort, rather than to any expertness of his own. Such success is of a very doubtful nature, and if not bordering on financial loss it is at least secured only at a low working efficiency.

As examples of good drawings the plates shown are not claimed to be anything extraordinary, but it will be noted that they are clean-cut and definite, and that even the sketches are unmistakable as to that which they are intended to illustrate. The information as to the design is all there; nothing is left to the imagination.

Classification of Machinery. It is intended to be made clear in all that has preceded, that the same method of attack and procedure may be applied to the designing of machinery, whatever may be the class or kind. This is a fundamental principle. When it is logically carried out, however, it produces very different results, as is evidenced by the characteristics of style peculiar to each of the classes of machinery to one or another of which all machines belong.

For example, an engine lathe has a style similar to a drill press, or a boring mill, or a screw machine, or a milling machine. It is very different, however, from the style of a steam engine, or a pump, or an air compressor, or a locomotive; it is still more different from the style of a rolling mill, or a link belt conveyor, or a coal crusher, or a stamp mill.

These classes of machinery are so distinctly marked that the novice is easily able to perceive that there is some controlling influence in each which marks its peculiar style. He should at the same time see that the very analysis that has been so strongly insisted upon in these pages is the direct cause of the marked characteristic in design. Each class of machinery must satisfy certain exacting conditions different from those of any other, and it is the careful study of these conditions, as fundamentally enforced, which leads to the strictly logical design.

A few of the most common classes are enumerated below, and their prominent features noted. It is hoped that a study of them will familiarize the student in a general way with the requirements of each, and serve as a guide to a more comprehensive study of their detail design than is possible in these pages.

Machine Tools. Examples:—lathe, planer, milling machine, drill press, screw machine, boring mill, grinding machine, etc., etc.

The machines of this class are all utilized for the finishing of metal surfaces. They are really at the root of the production of machinery of all other classes. Accuracy is their prime characteristic—accuracy of construction, accuracy of operation, accuracy of adjustment. Any inaccuracy that exists primarily in a machine tool is reproduced in every piece upon which it produces a finished surface; and since the mere act of finishing a surface upon anything implies that a rough and inaccurate surface will not answer, the tool then fails of its purpose if it cannot produce a true surface: it does not accomplish that for which it was designed.

The effect that this element of accuracy has upon the design of a machine tool is to require long bearings, convenient and exact methods of adjustment, stiffness, excess of material to absorb vibration, special shapes to facilitate application of jigs, fixtures, and exact manufacturing devices insuring interchangeability of parts, dust guards, and automatic lubrication.

Machine tools are essentially machines of maximum output, and depend for their success, not only upon their accuracy as noted, but also upon their ability to do the greatest amount of work per square foot of space occupied, with the least amount of manual labor and attention on the part of the operator. This is especially true of automatic machinery, which perhaps might be classed by itself in this respect, but which is nevertheless included under the broad term of a machine for producing finished surfaces, being merely the highest and most refined form of same. For machines of this class the designer has to study every detail with the most minute attention, packing away the operating parts into the smallest space and yet providing ready means for access, removal, and repair. Clearances that would be too little for other kinds of machinery are permitted and provided for; material of high grade, strength, and wearing quality, though expensive in first cost, and requiring the most expert skill to finish and to fit into place, must be used in order to keep the machine compact and yet of large capacity, to make it reasonably light in weight and yet amply strong.

Another point which has a great influence on the design of a machine tool is that we can never tell in advance just what it will have to stand in work, for the variation in the material that it finishes, the uncertain skill of the operator who runs it, the crowding to its limit of capacity and even beyond in times of press of business, and the many other stresses that may suddenly and without warning be thrown upon it, must all be thought of and provided for.

The points above mentioned are but a few of those which the designer of machine tools has to meet, and are presented merely as illustrations to show the special skill required in this class of machinery. It is readily seen that while the machine tool designer has great latitude in choice of material and in expenditure of money for refinement of structure—perhaps greater latitude than in any other class, yet he is held down as in no other to the final productive results, a small percentage of failure entirely throwing out the machine as a marketable product.

The style and external appearance of machine tools have a character of their own resulting from this extreme detailed care in design. Corners and fillets are carefully rounded; surfaces and intersections are definitely made; in short, the mechanical beauty of a machine tool is seen only from a near view and close inspection, and it is to this end that the design is constantly directed Appearance is a large factor in the sale of a fine tool, and the prestige of the American trade abroad in this respect is very noticeable.

Motive-Power Machinery. Examples:—Steam engine, gas engine, air compressor, steam pump, hydraulic machinery, etc., etc.

The element of heat enters into the design of all machinery in this class. The natural agents, air, gas, and water, in their various forms, are taken into the machine in the most efficient form in which it is possible to obtain them, are robbed of their energy to provide power, and are discharged in a form as weak and inert as the efficiency of the machine will determine.

In contrast to the class of machinery just studied, it should be noted that these machines do not produce any material thing; that is, they do not produce finished surfaces on metals, make screws or bolts, bore holes in castings, or turn line shafting. They merely take the energy of the natural agent, which is not in a form available for use, and transform it into motive power for general use.

Hence the element of accuracy as entering into the design of these machines is necessary only for their own efficient operation, and not for the quality of the thing which they produce, as in the case of machine tools. For example, the power furnished by one steam engine to drive a line shaft is as good as that of another as far as the rotating of the shaft is concerned, provided, of course, that both are equipped with the same quality of governing mechanism. The fact that one of the engines has a good adjusting device on the main bearing while the other has not is of no consequence from the standpoint of the line shaft, but it is, of course, of consequence respecting the efficient operation of the engines.

The design of steam engines and similar machines is of a rough nature compared with that of machine tools, as far as the detail of surface is concerned. General accuracy is nevertheless essential for the machine's own sake, but while in the machine tool we deal with thousandths of an inch, in the steam engine hundredths of an inch indicates fine work.

These machines are subject to extremes of temperature that have to be provided for in the design and arrangement of the parts. Being prime movers, controlling the operation of many machines, they must be certain to run during their period of work; hence design and adjustment must be positive, and when the latter cannot be made while running, it must be quickly and definitely accomplished when a stop is made. Simplicity of construction is essential, facilitating cheap and quick repairs. The design should be such that constant attention while running is avoided, the usual attention of the engineer being a safeguard rather than an implied factor of the original design. General rigidity and stiffness are important, also good balancing of the moving parts, and weight for absorption of vibration; otherwise under the constant daily run the machines will tear to pieces not only themselves but their foundations.

As far as external appearance goes in this and subsequent classes to be mentioned we are on a very different basis from that of machine tools. *General* mechanical symmetry of form is aimed at in the design, and the several smaller parts depend for their outline (aside from considerations of strength, which are, of course, always in order) upon the harmonious relation which they bear to the main and fundamental elements of the machine. Such machinery as air compressors, steam engines, pumps, and the like are viewed as a whole, and criticised, not detail by detail, as is the machine tool, but as to general effect of outline observed from some distance. To convey the desired effect to the eye the design must be bold and massive, connections simple and direct, and the smaller parts must not be so dwarfed in size as to appear like delicate ornaments instead of integral parts of the machine. The lines of connected parts must be continuous from one part to the other; and when interrupted by flanges, bosses, or lugs, the latter, which are merely incidental to the former must not be allowed to obscure wholly the main lines of the fundamental pieces.

It is attention to such points as these that marks the difference between well-designed motive-power machinery and that of the opposite character. Even though the little details of fillets and corners and surfaces may have their effect from a close point of view, the design will stand or fall in excellence on its bolder features, as noted above.

Structural Machinery. Examples:—Hoists, cranes, elevators, transfer tables, locomotives, cars, conveyors, cable-ways, etc., etc.

In the two preceding classes that have been noted, cast iron in the form of foundry castings enters as the principal material. Steel is utilized for shafts, studs, pins, and keys. Also special forgings, malleable iron and steel castings enter as factors in the production of the machinery discussed. Foundry castings, however, compose the great body of the material used, and the chief problems involved are those of the expert moulding of cast iron, and the handling and finishing of the same. For the operating parts, steel of fine grade is used in highly finished form, expensive because of its fineness, and yet a necessity to the extent it is used. Brass and bronze are used in the same way, generally in connection with the bearings for the shafts.

Structural machinery, on the contrary, uses steel as the basis of its construction. The fundamental structure is built up of plates, channels, beams, and angles; castings, though numerous, are relatively small, being riveted or bolted to the main structure and controlled in their design by its requirements.

Steel is used in this manner partly because the exclusive use of castings is prohibited on account of the excessive weight, and therefore expense, and partly because castings could not be made which would possess the necessary toughness and strength. In many cases the size of the machinery is such that castings, even if they could be made, would not support their own weight. Moreover, machinery of this class is subjected to rough service, and yet must be practically infallible under all conditions, neither being uncertain in operation at critical moments nor entirely failing under an extraordinary load.

The design of structural machinery is tied up to conditions existing largely outside of the locality in which the machinery is built. The steel plates and structural shapes required, being products of the rolling mill, have to conform to the latter's standards. The rivets, bolts and other fastenings have to be in accordance with the established practice of the structural iron worker, in order to permit punching, shearing and bending machinery of regular form to be utilized. Shipment on standard railway cars has to be considered, the design often requiring to be modified to permit this and nevertheless insure positive and accurate assembling in the field.

Steel castings, both large and small, find ready application in this class of work; also steel forgings, requiring to be worked under a heavy hammer and in many cases by specially devised processes.

In structural design less of the actual process of manufacture is under the eye of the designer than in the former classes of machinery which have been considered, and hence more allowance has to be made for things not coming exactly right to the fraction of an inch. It would be bad design to plan any structural piece of work with the same closeness of detail permitted, and in fact required, in the case of machine tools, or even in the case of motive-power machinery. In planning structural work the idea must be carried out, of certainty of operation in spite of roughness of detail and variations of construction. This does not necessarily imply inaccuracy, or shiftless, loosely constructed machinery; on the contrary, quite the reverse. The locomotive, for example, is one of the most refined pieces of mechanism that exists today; and yet the methods applied to the construction of machine tools would prove a failure on the locomotive. The design of a car axle box has to be just right else it will heat and destroy itself; the same is true of the spindle of a fine engine lathe; and yet how rough the former is compared with the latter, and how unsuited either would be for use on the service of the other.

As a general rule structural machinery can be more closely proportioned to theoretically calculated size than can the preceding types. The rolled material of which it is made is of a uniform and homogeneous nature owing to its process of manufacture, hence its every fibre may be counted on to sustain its share of the total load imposed upon it. This is in sharp contrast to the case of cast iron, which is of such a porous and irregular structure that we have to use a large factor of safety to cover this inherent defect.

Steel castings of both small and large size (which are quite apt to be utilized in this class of machinery for parts that can with difficulty be made out of rolled material), if properly designed of uniform thickness, with all corners well filleted and with the channels for the flow of the molten metal direct and ample, are nearly as reliable as rolled steel. In parts subject to excessive vibration, shocks, and sudden wrenchings, as, for example, the side frames or the connecting rod of a locomotive, the forged and hammered material is practically a necessity. This is especially the case when the possible breakage of the part would cause serious consequences involving heavy loss of life and property.

From the several points of view as above considered, it can be readily appreciated that, while structural work is in one sense rough and unpolished, yet it requires, from an engineering standpoint, quite as much breadth of experience and judgment as any of the other types. The fine-tool designer, least of all, perhaps, requires book theory, but does require an extended machine-shop experience. The designer of motive-power machinery needs pure physical theory and shop experience of a large and broad scope. The structural designer is least of all concerned with refined and minute finishing processes, but utilizes his theory absolutely, even though roughly.

Mill and Plant Machinery. Examples:—Rolling mills, mining machinery, crushers, stamps, rock drills, coal cutters, the machinery of blast furnaces and steel mills, tube mills, etc., etc.

This machinery constitutes a class which in the roughness of its operation exceeds all others. Moreover, it is machinery which for the most part is in continuous operation—24 hours per day and 365 days in the year. Hence refinement, even such as might be permitted in the preceding class of Structural Machinery, would be fatal here. The conditions that surround plant machinery are unfavorable in the extreme to the life of any material or metal, and it is not possible to change these conditions or give more than partial protection to the operating parts. Hence the design of such machinery must proceed primarily on the assumption that abuse and neglect, grinding away of surfaces, chemical eating away of metal, flooding of parts with water gritty and corrosive, subjection to sudden bursts of flame and intense heat, etc., will in a relatively short time totally destroy, perhaps, the entire structure.

In view of the continuous nature of the working process, which must be kept up in spite of these almost insurmountable conditions, the problem in each case becomes one of expediency; and the designs and arrangement of machinery must be so worked out that operation, repair, construction, and installation can all go on simultaneously without stopping the continuous process, and with but a small degree of inconvenience to the operation of the plant.

This problem, difficult though it may seem, can be worked out successfully, as is evidenced by the great number of plants of the continuous character operating at high efficiency throughout the world. The engineering and designing skill required to accomplish this, is perhaps of the highest degree met with in modern practice, for in it is involved a working knowledge of the possibilities, if not the detailed designs of machinery included in all classes. And yet, as in the most elementary case of simple design that can be conceived, the result is accomplished in the same way, namely, by studying the conditions (analysis), developing an ideal application to those conditions (theory), and then reducing the ideal design to a practical basis (modification).

A Few Pointed Suggestions on Original Design. Original design deals with the development of original mechanical ideas. The prime requisite for the development of an idea is to understand thoroughly the idea in the rough. See distinctly the mark aimed at, and never lose sight of it. If a method of reaching it is already outlined, understand that also thoroughly and the principles involved. It is impossible to go ahead blindly and hope to come out right. No good machine was ever built that does not stand for hours of concentrated thought on the part of its designer. Good machines never *happen*, they always *grow*.

Just as soon as the object to be accomplished is clearly understood, begin to produce some visible work on the problem. Sketch something. Get some ideas on paper. Ideas on paper suggest other ideas. If the problem, for example, is one of lathe design, sketch a rectangle, and call it the headstock; another rectangle, and call it the footstock; a couple of scratches for the centers; some steps for the cone pulley; three or four lines for the bed; and as many more for the supports. There is now something on paper to look at; the design is begun.

It is much better to stare at this sketch, than into blank space trying to imagine the finished design. No matter how rough the sketch may be, a short study of it will develop some limiting conditions that before were not apparent. Guess at a few rough dimensions; put them on the sketch; develop another view—a plan or a side elevation—all still in the roughest style, without any regard to finished detail. Information will be growing all the while, and the problem will be opening up. At this stage it is probable that the sketch can easily be seen to be wrong in many respects. Perhaps the arrangement will not do at all.

This is a good sign. It shows that the design is progressing. It is a valuable thing to know that certain plans *cannot* be followed. Do not rub out part of the sketch already made and try to remedy it. Begin again. Make another sketch. Sketch paper is cheap. By and by it may prove to be very desirable to have that first rough outline available for comparison; or it may be that some of its ideas can be applied on other sketches. The second sketch may "show up" little or no better than the first. Make another, and another, and another, until the subject is thoroughly digested. It is wonderful how helpful it is to have some marks on paper relative to a design, even though they be of the utmost crudeness. They save imaginative power tremendously; and, even with them, all available powers of imagination will be needed before the design is perfected.

A careful comparison of one's sketches, rejecting here, and approving there, will, little by little, bring about a definite opinion, and the scale drawing can be begun.

As in the case of the first sketch, so in the case of the first scale drawing, get some lines on paper as quickly as possible. Draw something, even if it is nothing more than a straight horizontal line. Do not stare at blank paper for an hour trying to imagine how the tenth or eleventh line is going to be drawn in relation to the first line. Do not worry about the later lines until it is time for them Draw the first line at once; and, when the second line is drawn, if the first line proves to be wrong, make it right. As in the rough sketch, that first horizontal line is an immense relief from the great waste of blank paper of a fresh sheet. It is something to look at. It is the beginning of a detailed design. If it happens not to be the absolutely correct foundation to build upon, it at least is something to tear down. The main purpose of these preliminary drawings is to keep the mind active on the problem; and advance toward the final accomplishment of the design is often made quite as rapidly by discovering what to tear down as by consistently building up.

When a detail draftsman who has been used to having all his work laid out for him by an expert designer attempts to take up original work for himself, he encounters the drawing of that first line in a way he never did before. He is apt to worry for some time over the possible or impossible results of drawing that first line. If he continue this, he will be sure to fail. The second line is much easier to draw than the first, and the third than the second; and the next hundred will follow on in comparatively smooth sequence, all because of bold action on the first few lines.

And yet, just as the design appears to be progressing smoothly, and the advanced progress of the drawing seems cause for congratulation, careful consideration may disclose a "snag" not previously known to exist in the problem. Further study pursued along the line of this new discovery may show that the whole layout thus far has been radically wrong, and that a fresh start will have to be made. At such a time the young designer is apt to feel that his labor has all been thrown away, and he becomes discouraged. There is, however, no cause for discouragement. Machine Design might almost be defined to be the "successful elimination of snags." It takes some ability to discover an obstacle of this sort; to know a "snag" when an opportunity to see it is given. It takes a good designer to eliminate such a difficulty after it has been found. If there were no "snags" it would not require great ability to design machines. Many machines fail because in them there are a lot of undiscovered "snags." Others fail because the "snags," although discovered, were not eliminated by careful design.

Do not be afraid to make a lot of "first" drawings. It is just as important to digest the design thoroughly by means of scale drawings, as it was to digest it originally by means of the rough sketches. An attempt to make the first drawing of an original design absolutely right would, it is safe to say, produce a poor design, one that could be much improved by further trial. Let the drawings multiply, one after another, until the final one is reached, in which the perfection of detail will eliminate all the bad points of the preceding drafts and incorporate good ones of its own based on the study of the others.

And yet it is often true that the first design laid out, even after many others have been developed, may be found to possess features that render a return to it desirable. This is why it is always better to produce a collection of designs than to attempt to rub out and work over the first one. The best designers usually have a great number of sketches showing how to accomplish a single result. Likewise, they also have a series of layouts to scale, showing in detailed form the development of their various ideas. This is because, without a careful consideration of many methods, they themselves feel incompetent to judge of the best design possible for accomplishing a given result.

Sketches and original designs should always be dated and signed. Different designers may be working on the same problem, and priority of design will never be allowed except upon signed and witnessed papers. It is embarrassing to find, after months and perhaps years have passed since an original drawing was made, that one's rights have been preëmpted merely because there was no date or signature to define them.

In redesigning or modifying an existing machine, never make a change merely for the sake of doing so. Give the good points of the machine a chance, and devote attention in the new design to correcting the bad points. It is in bad taste, if it be not actually childish, to "look wise and suggest a change" in details which happen to have been designed by another party, but which, nevertheless, are by common engineering judgment pronounced good for the special work intended. This element of unfair and selfish criticism has more than a moral bearing. When it is carried into the superintendence of designing work, it extinguishes the personality of the subordinate draftsman; his efficiency as an original thinker is lowered; and narrow designs are produced.

"The best way for a subordinate to dispose of what appears to be a poor suggestion from a superior, is to work it out to the best degree possible." If it turns out to be good the credit of working it out belongs to the man who did it. If it is actually bad, a careful working out will usually develop the fact beyond dispute, and save unprofitable argument. For the success or failure of a machine there is only one argument better than the detail drawings, and that is the machine itself in operation.

Detail drawings, however, are infinitely better prosecutors or defendants than a multitude of wordy counsel.

Summary. The above classification of machinery might be subdivided and extended indefinitely, and on the broad basis on which it is given it doubtless does not cover the entire field. As an illustration, however, not only of types of machinery, but of methods of design and study, it is hoped that it may be of assistance in giving a start to the student of machine design, in whatever class his interests may happen to lie.

It is the general principles of the art which it is important to master. It is not the designing of a locomotive, or a stationary steam engine, or a crane, or an engine lathe, or a rolling mill, which should be sought to be learned, but the designing of *anything* that may confront us. Specializing is sure to come to the designer in the course of his experience, and when it does he merely fits to the particular specialty the principles he knows for all, and practically develops them along that individual line.

EXAMINATION PAPER

MACHINE DESIGN.

Instructions to the Student. Place your name and full address at the head of the paper. Work out in full the examples and problems, showing each step in the work. Mark your answers plainly "Ans." Avoid crowding your work, as this leads to errors and shows bad taste. For the calculations any cheap, light paper like the sample previously sent you may be used.

The drawings made in accordance with the problems below should be traced in ink on tracing cloth 18 by 24 inches in size, and having a border line $\frac{1}{2}$ inch inside the edge of the paper.

PROBLEMS.

1. Suppose a 30-inch pulley is substituted for the 42-inch in the problem given, and that the pulley on the motor remains $10\frac{1}{2}$ inches as before, how fast must the motor run to give the rope the same speed, 150 feet per minute?

2. Will the horse-power of the motor be changed with this new condition? Explain fully.

3. Calculate the width of double belt for above condition.

4. What is the torque on the motor shaft for above condition?

5. Calculate the size of shaft in the small pulley for above condition.

6. Calculate the size of shaft in the 30-inch pulley for above condition.

7. Design and draw both pulleys for above condition, making complete working drawings, and giving all calculations in full.

8. Taking the original problem as given in the text, suppose it is desired to increase the large gear to 45 inches diameter, calculate the load on the tooth, and a suitable pitch and face to take this load.

9. How many teeth must the pinion have to give the same speed of rope, 150 feet per minute, assuming that the motor runs 470 revolutions per minute, for condition in Problem 8?

10. Calculate the bore of pinion for this case.

11. Design and draw the gears for the conditions of Problems 8 and 9, giving all calculations in full.

12. When there is but 3,000 pounds on the rope, what are the tensions in each end of the brake strap, assuming that the size of drum and other conditions remain the same?

13. How much pressure on the foot lever would it take to hold this load of 3,000 pounds on the rope?

14. Suppose we put a bearing 9 inches long on the drum shaft; the distance, center to center of bearings, would then be 3 feet $5\frac{3}{4}$ inches, gears, drum, brake, and load being same as in the original problem of the text. Calculate the diameter of the drum shaft.

15. Suppose the height of bracket, center to base, to be 15 inches; length and diameter of bearing, as in Problem 14; and that we use a separate bracket for the drum bearings, not connected with the pinion-shaft bearings. Design and draw such a bracket.

16. Calculate, design, and draw all the parts for a machine similar to that of the text, from the following data:

Load on rope	4,000 pounds.
Speed of rope	175 feet per minute.
Length of rope to be reeled in	250 feet.

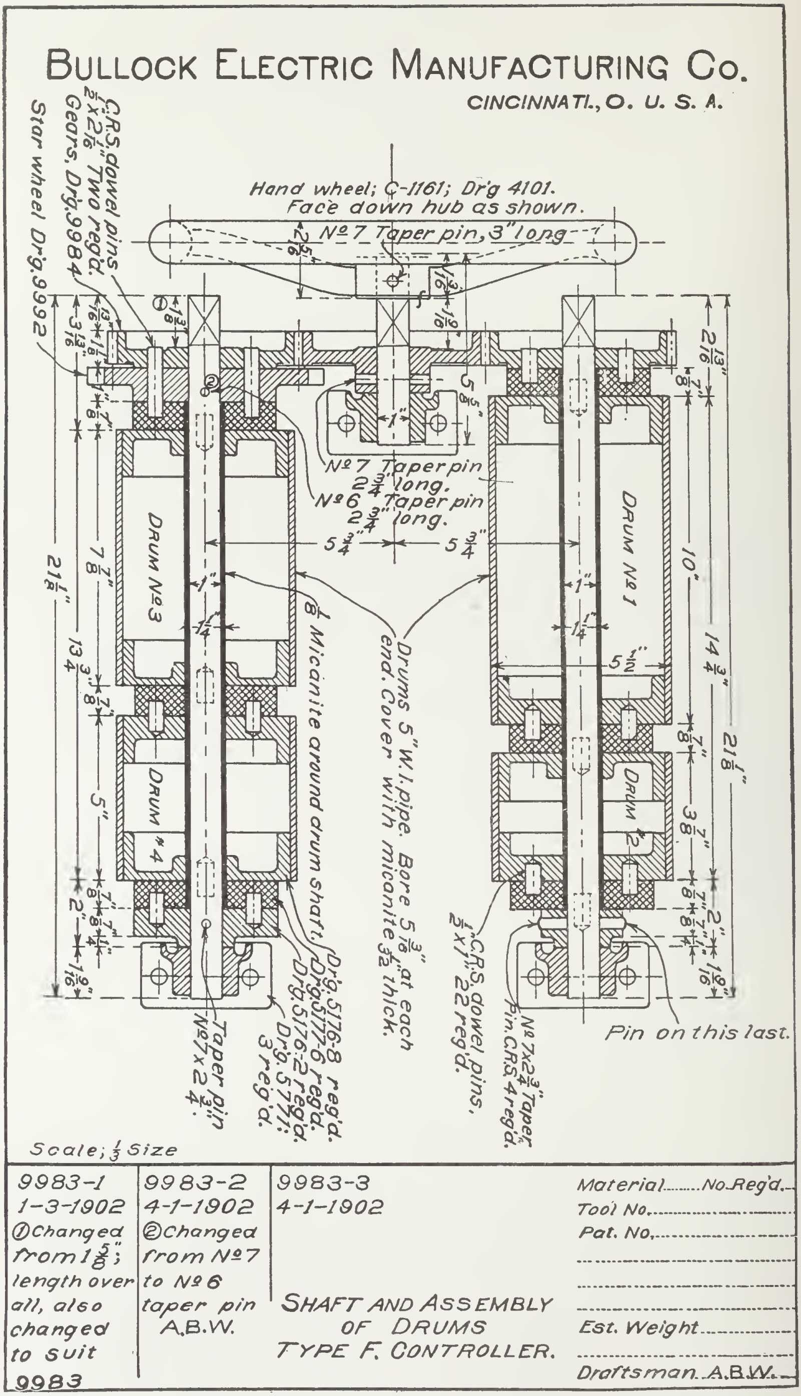
Note. Of the above problems, 12, 13, 15 and 16 are optional. Problems 12, 13 and 15 are comparatively simple, following closely the steps of the text in their solution.

Problem 16, likewise, is supposed to be worked out on the same lines as the text, but is wholly original in its nature, being based on entirely new data. It is not expected that this problem will be attempted except by well-advanced students who can give considerable time to working it out completely. It will be found, however, an excellent exercise in original and yet simple design.

After completing the work, add and sign the following statement:

I hereby certify that the above work, including accompanying drawings, is entirely my own.

(Signed)


Bullock Electric Manufacturing Co.
Cincinnati, O. U. S. A.
Hand wheel; C-1161; Dr'g 4101.
Face down hub as shown.
Nº 7 Taper pin, 3" long
C.R.S. dowel pins 1/2"x2 1/16" Two req'd.
Gears, Drg. 9984
Star wheel Drg. 9992
Nº 7 Taper pin 2 3/4" long.
Nº 6 Taper pin 2 3/4" long.
5 3/4"
5 3/4"
Drum Nº 3
Drum #4
Drum Nº 1
Drum #2
1"
1 1/4"
5 1/2"
1/8 Micanite around drum shaft.
Drums 5" W.I. pipe. Bore 5 3/16" at each end. Cover with micanite 1/32 thick.
Drg. 5176-8 req'd.
Drg. 5177-6 req'd.
Drg. 5176-2 req'd.
Drg. 5771; 3 req'd.
Taper pin Nº 7 x 2 3/4".
1/2" C.R.S. dowel pins, 1/2x1" 22 req'd.
Nº 7 x 2 3/4" Taper pin. C.R.S. 4 req'd.
Pin on this last.
21 1/8"
13 3/4"
7 7/8"
5"
2"
10"
14 3/4"
3 7/8"
2 13/16"
Scale; 1/3 Size
9983-1
1-3-1902
① Changed from 1 5/8"; length over all, also changed to suit
9983
9983-2
4-1-1902
② Changed from Nº 7 to Nº 6 taper pin
A.B.W.
9983-3
4-1-1902
Shaft and Assembly of Drums Type F. Controller.
Material........No. Req'd.
Tool No.
Pat. No.
Est. Weight
Draftsman A.B.W.

EXAMPLE OF SHOP DRAWING REPRODUCED FROM A BLUE PRINT BY THE COURTESY OF THE BULLOCK ELECTRIC MFG. CO.

MACHINE DESIGN.

PART II.

Introduction. In Part I is illustrated a definite and systematic method of attacking the design of a *machine as a whole.* In Part II the same plan is followed with regard to the detail of its *component parts*, the machine elements which are chosen as illustrations of the method, being the simplest and most familiar forms in common use.

As before, the student must strive to grasp and absorb the *method* of design rather than any specific and established form of a machine part. Part II is not a compendium of design, does not attempt to be complete or exhaustive in any of its chapters, but is condensed and simplified in order to lead the student into systematic mechanical thinking and logical and definite action. Each chapter is intended to stimulate to further and more exhaustive study along lines broader than, and under conditions different from those that can be specified in a general discussion. But no matter how deeply investigation may be carried, or how specialized the study may become, the student must realize that his path of action in any case whatsoever must lie along the lines of *Analysis*, *Theory*, and *Practical Modification* systematically applied.

BELTS.

NOTATION—The following notation is used throughout the chapter on Belts:

A = Sectional area of belt (square inches) $= bh$.
b = Width of belt (inches).
F = Force of friction at pulley rim (lbs.).
h = Thickness of belt (inches)
μ = Coefficient of friction.
N = Number of revolutions of pulley per minute.
n = Fraction of circumference of pulley embraced by belt.
P = Driving force at pulley rim (lbs.) $= F$.
R = Radius of pulley (feet).
r = Radius of pulley (inches).
T = Initial tension (lbs.).
T_n = Total tension on tight side (lbs.).
T_o = Total tension on slack side (lbs.).
t = Working tension of belt (lbs. per square inch).
V = Velocity of belt (feet per minute).
w = Weight of belt per cubic inch (lbs.).
z = Factor due to centrifugal force.

ANALYSIS. When a belt is stretched over a pair of pulleys, is cut off at the proper length, and is laced together into an endless band, it is evident that as long as the belt is at rest there is a nearly uniform tension in it throughout its length, due to the tightness with which the lacing is drawn up. If the distance between the pulleys is considerable, the weight of the belt itself as it hangs between the pulleys will produce a slightly greater tension next to the pulleys than exists in the middle of the span. This increase of tension due to the weight of the belt would make but little difference in the unit-stress in the material of which the belt is made; hence it may safely be assumed that the tension in the belt when at rest is uniform throughout its entire length.

When we start to transmit power through the belt by turning one of the pulleys, thereby driving the other pulley the condition of stress in the belt is at once materially changed. As the belt is a flexible member, we can transmit only a pull to the other pulley, thereby turning it around, the push which is at the same time given to the other side of the belt merely acting to make the belt sag or become slack. Hence the immediate effect of starting motion in a belt is to change the condition of equal tension throughout its length, to that of unequal tension in the two sides. The driving side is tight, while the other is loose, the former having gained as much tension as the latter has lost, and the sum of the two being practically equal to the sum of the tensions in the two sides of the belt when at rest. This is not strictly true, as will be shown later; but it is sufficiently accurate to form a good basis for the practical design, at least of slow-speed belts.

This condition of tight and slack sides is made possible by the fact that the belt, in being wrapped around the pulleys under tension, has friction on their surfaces. Thus, we can pull hard on one side without slipping the belt around the pulleys, but could not do this if the pulleys were perfectly smooth or frictionless, for in that case the slightest pull on one side would slip the belt around the pulleys. In fact, it would be impossible to produce any pull by means of the driving pulley, for the pulley would merely slip around inside the belt.

The amount of pull we can apply to the belt is therefore limited by the tension at which the belt slips around the pulley. Moreover, since the force of friction between the belt and pulley is dependent upon the normal force with which the belt is pressed against the pulley, and the coefficient of friction between the two, it is evident that the tighter the belt is laced up, and the rougher the surfaces of the pulley and belt, the greater is the force that can be transmitted through the belt. This leads to the conclusion that it would be possible to transmit any amount of power through any belt however small, if the belt were only laced up tight enough.

This conclusion is literally true; but the important fact now comes in, that the strength of the material of which the belt is made is limited, and while theoretically we might be able to accomplish the above, it would be impossible to do so in practice, for at a certain point the belt would break under the strain. Other practical considerations also come in, which fix this limit of power transmission at a point far below the breaking strength of the material.

The complete analysis is not quite as simple as the above, especially for high-speed belts. When the driving side of the belt becomes tight, it stretches and grows longer; and at the same time the other side of the belt becomes slack and grows shorter. But it is not true that the increase in the one side is the same as the decrease in the other, and this fact produces the condition that the sum of the tensions in motion is not quite the same as the sum of the tensions at rest.

Again, when the belt, as it passes around the pulley, changes its straight-line direction to circular motion, each particle of the belt—like a body whirling at the end of a cord about a center of rotation—tends by centrifugal force to fly away from the surface of the pulley, thereby decreasing the normal pressure, and hence the friction. This centrifugal force also changes somewhat the tensions in the belt between the pulleys. As the centrifugal force increases in proportion to the square of the linear velocity, it is evident that the effect is greater at high speeds than at moderate or low speeds.

A further circumstance that affects the driving power of a belt is the stiffness of the leather or other material of which the belt is made. As it passes around the pulley, the belt is bent to conform to the circumference of the pulley, and is again straightened out as it leaves the pulley. Hence the theoretically perfect action is modified somewhat according to the sharpness of the bending and the thickness or flexibility of the belt; in other words, a small pulley carrying a thick belt would be the worst case for successful calculation on a theoretical basis.

THEORY. The condition of the tight and loose sides of a belt transmitting power, is similar to that of the weighted strap and fixed pulley shown in Fig. 17. If motion is desired of the strap around the pulley, it is necessary to make the weight W_2 of such a magnitude that it will overcome not only the weight W_1, but also the friction between the strap and the pulley. The strap tension T_n is, of course, equal to W_2, and T_o to W_1. The equation showing the balance of forces for the condition when motion is about to occur, is:

$$T_n - T_o = F = P \text{ (driving force).} \qquad \textbf{(5)}$$

If the pulley be free to turn on its axis, instead of being fixed as in Fig. 17, the strap by its friction on the pulley will turn the pulley, and the force of friction F becomes the driving force for the pulley as noted in equation 5 above.

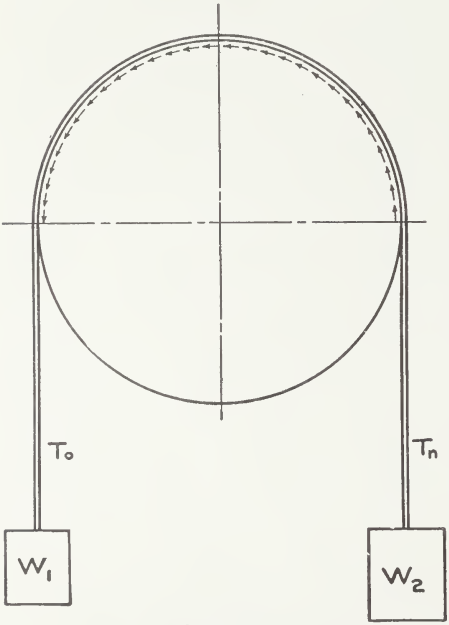


Fig. 17.

In Fig. 18, let us suppose that W is a weight representing the resistance to be overcome. The tensions T_n and T_o, equal at first owing to stretching the belt tightly over the pulleys at rest, change when an attempt is made to raise the weight by turning the larger pulley; and just as the weight leaves the floor, the equality of moments about the axis of the driven pulley gives the following equation:

$$(T_n - T_o)\, r = F \times r = P \times r = W \times r_1. \qquad \textbf{(6)}$$

This equality of moments remains as long as the motion of the weight is uniform, and represents closely the conditions under which belt pulleys work.

Although we know from the above what the *difference* of the belt tensions is, and what this difference will do when applied to the surface of a given pulley, we do not yet know what either T_n or T_o actually is; and until we do know, we cannot correctly proportion the belt. Hence we must find another relation between T_n and T_o which we can combine with equations 5 and 6. This relation is deduced by a process of higher mathematics, which results as follows:

$$\text{Common logarithm } \frac{T_n}{T_o} = 2.729\ \mu\ (1 - z)n. \qquad (7)$$

Treating equations 5 and 7 as simultaneous, values of both T_n and T_o can be found by the regular algebraic solution. As T_n is the larger, the actual area of belt to provide the necessary strength must be made to depend upon it.

The factor z in equation 7 depends upon the centrifugal force developed by the weight of the belt passing around the pulley. Its value, found from mechanics, is:

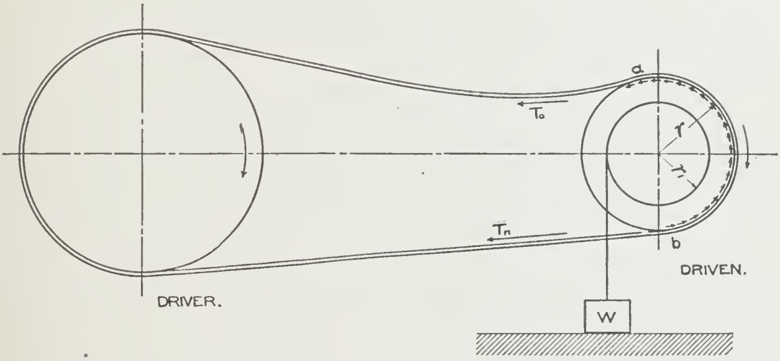


Fig. 18.

$$z = \frac{w \times V^2}{9{,}660 \times t}.$$

Having found the maximum pull on the belt, it now remains to write the equation:

$$\text{External force} = \text{Internal resistance};$$

or,
$$T_n = b \times h \times t. \qquad (8)$$

Usually the most convenient way to handle this equation is to assume h and t, and then solve for b.

Summing up the theoretical treatment of belt design, we simply combine equations 5, 6, 7, and 8, and solve for the quantity desired. Discussion of the constants involved in these equations, and of the practical factors controlling them, is given in the following:

PRACTICAL MODIFICATION. The force of friction F, which is the same as driving force P, depends on:

Coefficient of friction (μ) between belt and pulley;

Tightness of the belt;

Centrifugal force of the belt;

Angle of contact of belt with pulley.

The coefficient of friction (μ), according to experiments and observed operation of belts transmitting power, varies from .15 to .56 for leather on cast iron. An average value consistent with a reasonable amount of slip, the belt being in good running order, is .30. If the belt is oily, or likely to become so in use, a lower value should be taken.

The tighter the belt is drawn up, the greater is the pressure against the pulley, and hence the greater is the force of friction. But if we pull the belt up too tightly, when we begin to drive, T_n becomes too great, and the belt breaks or is under such stress that it wears out quickly. Moreover, the great side pressure on the bearings carrying the shaft produces excessive friction, and the drive is inefficient. This is why a narrow belt driven at high speed is more efficient than a wide belt at slow speed, for we cannot pull up the former as tightly as the latter without overstraining it, and yet it is possible to get the required power out of the narrow belt by running it at high speed.

The centrifugal force is of small importance for low speeds, say of 3,000 feet per minute and less; and it therefore may usually be neglected. The factor z then becomes zero in the expression $1 - z$ in equation 7, and the second member of the equation stands simply $2.729 \times \mu \times n$.

The angle of contact of belt with pulley is important, as a large value gives a great difference between T_n and T_o; and it is desirable to make this difference as great as possible, because thereby the driving force is increased. The loose side of a horizontal belt should always be above, as then the natural sag of the loose side due to its slackness tends to increase the angle of contact with the pulley, while the tightening up of the lower side acts against its sag to make the loss of wrap as little as possible. Vertical belts which have the driving pulley uppermost, utilize the weight of the belt to increase the pressure against the surface of the pulley, slightly increasing its capacity for driving. The angle of contact may be artificially increased by a tightening pulley which presses the belt further around the pulley than it would naturally lie. It adds however, the friction of its own bearing, and impairs the efficiency of the drive. For ordinary horizontal belts, the angle of contact is but little more than 180°, and the value of n in equation 7 may be safely assumed at $\frac{1}{2}$ unless the pulleys are of relatively great difference of diameter and very close together.

Strength of Leather Belting. The breaking tensile strength of leather belting varies from 3,000 to 5,000 pounds per square inch. Joints are made by lacing, by metal fasteners, or by cementing. The strength of a laced joint may be about $\frac{7}{10}$, of a metal-fastened joint, about $\frac{1}{2}$, and of a cemented joint, about equal to the full strength of the belt cross-sectional area. The proper working strength of belting depends on the use to which the belt is put. A continuously running belt should have a low tension in order to have long life and a minimum loss of time for repairs. For double leather belting it has been shown that a working tension of 240 pounds per square inch of sectional area gives an annual cost — for repairs, maintenance, and renewals — of 14 per cent of first cost. At 400 pounds working tension, the annual expense becomes 37 per cent of first cost. These results apply to belts running continuously; larger values may be used where the full load comes on but a short time, as in the case of dynamos.

Good average values for working tensions of leather belts are:

Cemented joints,	400	pounds per square inch.
Laced joints,	300	" " " "
Metal joints,	250	" " " "

Horse-Power Transmitted by Belting. If P is the driving force in pounds at the rim of the pulley, and V is the velocity of the belt in feet per minute, the theoretical horse-power transmitted is evidently :

$$\text{H. P.} = \frac{P \times V}{33{,}000}. \qquad \textbf{(9)}$$

It is evident from the above that the horse-power of a belt depends upon two things, the driving force P and the velocity V. If either of these factors is increased, the horse-power is increased. Increasing P means a tight belt. Hence a tight belt and high speed together give maximum horse-power. But a tight belt means more side strain on shaft and journal. Therefore, from the standpoint of efficiency, *use a narrow belt under low tension at as high a speed as possible.*

Empirical rules for horse-power of belting, if used with judgment, give safe results when applied to very general cases. A common rule used by American engineers is:

$$\text{H. P.} = \frac{b \times V}{1{,}000}. \qquad \textbf{(10)}$$

For a double belt, assuming double strength, this becomes:

$$\text{H. P.} = \frac{b \times V}{500}. \qquad \textbf{(11)}$$

With large pulleys and moderate velocities, this may hold good. With small pulleys and high velocities, however, the uncertain stresses induced by the bending of the fibers of the belt around the pulley, and the relatively great loss due to centrifugal force, modify this relation· and a safer value for a double belt of the ordinary kind is:

$$\text{H. P.} = \frac{b \times V}{540}; \qquad \textbf{(12)}$$

or, still safer,

$$\text{H. P.} = \frac{b \times V}{700}. \qquad \textbf{(13)}$$

If we compare the theoretical value of equation 9 with the empirical value of equation 10 by putting them equal to each other, thus:

$$\text{H. P.} = \frac{P \times V}{33{,}000} = \frac{b \times V}{1{,}000}.$$

and solve for P, we get :

$$P = 33b. \qquad (14)$$

This develops the fact that the empirical rule of equation 10 assumes a driving force of 33 pounds per inch of width of single belt.

Another way of expressing equation 10 is: A single belt will transmit one horse-power for every inch of width at a belt speed of 1,000 feet per minute.

Speed of Belting. The most economical speed is somewhere between 4,000 and 5,000 feet per minute. Above these values the life of the belt is shortened; also "flapping," "chasing," and centrifugal force cause considerable loss of power. The limit of speed with cast-iron pulleys is fixed at the safe limit for bursting of the rim, which may be taken at one mile per minute.

Material of Belting. Oak-tanned leather, made from the part of the hide which covers the back of the ox, gives the best results for leather belting. The thickness of the leather varies from .18 to .25 inch. It weighs from .03 to .04 pound per cubic inch. The average thickness of double leather belts may be taken as .33 inch, although a variation in thickness from $\frac{1}{4}$ inch to $\frac{7}{16}$ inch is not uncommon. Double leather belts may be ordered light, medium, or heavy.

In a single-thickness belt the grain or hair side should be next to the pulley, for the flesh side is the stronger and is therefore better able to resist the tensile stress due to bending set up where the belt makes and leaves contact with the pulley face. Double leather belts are made by cementing the flesh sides of two thicknesses of belt together, leaving the grain side exposed to surface wear.

Raw hide and semi-raw hide belts have a slightly higher coefficient of friction than ordinary tanned belts. They are useful in damp places. The strength of these belts is about one and one-half times that of tanned leather.

Cotton, cotton-leather, rubber, and leather link belting are some of the forms on the market, each of which is especially adapted to certain uses. For their weights and their tensile and working strengths consult the manufacturers' catalogues.

A prominent manufacturer's practice in regard to the sizes of leather belting will be found useful for comparison, and is indicated in the table on page 12.

Initial Tension in Belt. On the assumption that the sum of the tensions is unchanged, whether the belt be at rest or driving, we should have the following relation :

$$T_n + T_o = 2T;$$

whence,

$$T = \frac{T_n + T_o}{2}. \qquad (15)$$

This is not strictly true, however, as is stated in the "Analysis" of "Belts." It has been found that in a horizontal belt working at about 400 lbs. tension per square inch on the tight side, and having 2 per cent slip on cast-iron pulleys (*i. e.*, the surface of the driven pulley moving 2 per cent slower than that of the driver), the increase of the sum of the tensions when in motion over the sum of the tensions at rest, may be taken at about $\frac{1}{3}$ the value of the tensions at rest. Expressing this in the form of an equation

Sizes of Leather Belting.

WIDTH.	THICKNESS.	
	Single.	Double.
1 inch.	$\frac{5}{32}$ inch.	$\frac{5}{16}$ inch.
2 "	$\frac{3}{16}$ "	$\frac{5}{16}$ "
3 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
4 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
5 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
6 "	$\frac{7}{32}$ "	$\frac{3}{8}$ "
10 "	$\frac{5}{16}$ "	$\frac{3}{8}$ "
12 "	....	$\frac{3}{8}$ "
14 "	....	$\frac{13}{32}$ "
20 "	....	$\frac{7}{16}$ "

$$T_n + T_o = \frac{4}{3}(2T) = \frac{8 \times T}{3}.$$

$$T = \frac{3}{8}(T_n + T_o). \qquad (16)$$

The value of T thus found would be the pounds initial tension to which the belt should be pulled up when being laced, in order to produce T_n and T_o when driving.

This value is not of very great practical importance, as the proper tightness of belt is usually secured by trial, by tightening pulleys, by pulley adjustment (as in motor drives), or by shortening the belt from time to time as needed. It is worth noting, however, that for the most economical life of the belt it would be very desirable in every case to weigh the tension by a spring balance when giving the belt its initial tension. This, however, is not always easy or even feasible; hence it is a refinement with which good practice usually dispenses, except in the case of large and heavy belts.

PROBLEMS ON BELTS.

1. Determine the belt tensions in a laced belt transmitting 50 horse-power at a velocity of 3,500 feet per minute. Suppose that the arc of contact is 180°; weight of belt = .035 pound per cub. in.; and coefficient of friction 25 per cent.

2. What is the width of above belt if it is $\frac{3}{16}$ inch in thickness?

3. What initial tension must be placed on above belt?

4. The main drive pulley of a 120-horse-power water wheel is 6 feet in diameter. A cemented leather belt is to connect the main pulley to a 3-foot pulley on the line shafting in a mill. The horizontal distance between centers of shafting is 24 feet; coefficient of friction, 30 per cent; revolutions per minute of line shafting, 180. Design the belt for this drive.

5. An 8-inch double belt $\frac{3}{8}$ inch thick connects 2 pulleys of 30-inch and 20-inch diameter respectively. The horizontal distance between the centers is 12.5 feet. The coefficient of friction is 0.3, and the weight of belt per cubic inch is 0.035 pound. Working tension, 300 pounds per square inch. Speed of belt 5,000 feet per minute. Lower face of 30-inch pulley is the driving face. Required the H. P. which may be transmitted (theoretically).

6. Compare the theoretical horse-power in problem 5 with that obtained by the use of empirical formula.

PULLEYS.

NOTATION—The following notation is used throughout the chapter on Pulleys:

A =Area of rim (sq. in.).
a = " " arm (" ").
b =Center of pulley to center of belt (inches; practically equal to R).
C_1 =Total centrifugal force of rim (lbs.).
c =Distance from neutral axis to outer fiber (inches).
D =Diameter of pulley (inches).
D_1= " " hub (").
d_1 = " " bolt at root of thread (inches).
d =Diameter of bolt holes (inches).
g =Acceleration due to gravity (ft. per sec.).
h =Width of arm at any section (inches).
I =Moment of inertia.
L =Length of arm, center of belt to hub (inches).
L_1=Length of rim flange of split pulley (inches).
l =Length of hub (inches).
N =Number of arms.
n = " " rim bolts, each side.
P =Driving force of belt (lbs.).
P_1=Force at circumference of shaft (lbs.).
P_2=Force at circumference of hub (lbs.).
p =Stress in rim due to centrifugal force (lbs. per sq. in.).
R =Radius of pulley (inches).
S =Fiber stress (lbs. per sq. in.).
s =Fiber stress in flange (lbs. per sq. in.).
T =Thickness of web (inches).
t = " " rim (").
t_2 = " " " bolt flange (inches).
T_n=Tension of belt on tight side (lbs.).
T_o= " " " " loose " (").
v =Velocity of rim (ft. per sec.).
w =Weight of material (lbs. per cub. in.).

ANALYSIS. If a flexible band be wrapped *completely* about a pulley, and a heavy stress be put upon each end of the band, the rim of the pulley will tend to collapse just like a boiler tube with steam pressure on the outside of it. A compressive stress is induced which is very nearly evenly distributed over the cross-section of the rim, except at points where the arms are connected thereto. At these points the arms, acting like rigid posts, take this compressive stress. Now, a pulley never has a belt wrapped *completely* round it, the fraction of the circumference embraced by the belt being usually about $\frac{1}{2}$, and seldom, even with a tightener pulley, reaching $\frac{3}{4}$. Assuming the wrap to be $\frac{1}{2}$ the circumference, and that all the side pull of the belt comes on the rim, none being transmitted through the arms to the hub, we then have one-half of the rim pressed hard against the other half by a force equal to the resultant of the belt tensions, which, in this case, would be the sum of them. Dividing the pulley by a plane through its center and perpendicular to the belt, the cross-section of the rim cut by this plane has to take this compressive stress-

This analysis is satisfactory from an ideal standpoint only, for the intensity of stress due to the direct pull of the belt, with the usual practical proportions of rim, would be very small. Moreover, the element of speed has not been considered.

When the pulley is under speed, a set of conditions which complicates matters is introduced. The centrifugal force due to the weight of the rim and arms is no longer negligible, but has an important influence upon the design and material used. This centrifugal force acts against the effect of the belt wrap, tending to reduce the compressive stress, or, overcoming the latter entirely, sets up a tensional stress both in the rim and in the arms. It also tends to distort the rim from a true circle by bowing out the rim between the arms, thus producing a bending moment in the rim, maximum at the points where the rim joins each arm.

It can readily be imagined that the analysis in detail of these various stresses in the rim acting in conjunction with each other is quite complicated — far too much so in fact, to be introduced here. As in most cases of such design, however, one controlling influence can be separated out from the others, and the design based thereon with sufficient margin of strength to satisfy the more obscure conditions. This is rational treatment, and the "theory" will be studied accordingly.

The rim, being fastened to the ends of the arms, tends, when driving, to be sheared off, the resisting area being the areas of the cross-sections of the arms at their point of joining the rim. The force that produces this shearing tendency is the driving force of the belt, or the difference between the tensions of the tight and loose sides.

Again, at the point of connection of the arms to the hub, a shearing action takes place, so that, if this shearing tendency were carried to rupture, the hub would literally be torn out of the arms. Now, viewing the arms as beams loaded at the end with the driving force of the belt, and fixed at the hub, a heavy bending stress is set up, which is maximum at the point of connection to the hub. If the rim were stiff enough to distribute this driving force equally between the arms, each arm would take its proportional share of the load. The rim, however, is quite thin and flexible; and it is not safe to assume this perfect distribution. It is usual to consider that one-half the whole number of arms take the full driving force.

THEORY—Pulley Rim. Evidently it is practically impossible to make so thin a rim that it will collapse under the pull of a belt. As far as the *theory* of the rim is concerned, its proportion probably depends more upon the calculation for centrifugal force than upon anything else.

In order to separate this action from that of any other forces, let us suppose that the rim is entirely free from the arms and hub, and is rotating about its center. Every particle, by centrifugal force, tends to fly radially outward from the center. This condition is represented in Fig. 19. The tendency with which one-half of the rim tends to fly apart from the other is indicated by the force C_1; and the relation between C_1 and the small radial force c for each unit-length of rim can readily be found from the principles of mechanics. The case is exactly like that of a boiler or a thin pipe subjected to uniform internal pressure, which, if carried to rupture, would split the rim along a longitudinal seam.

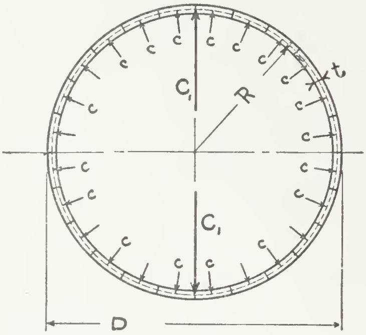


Fig. 19.

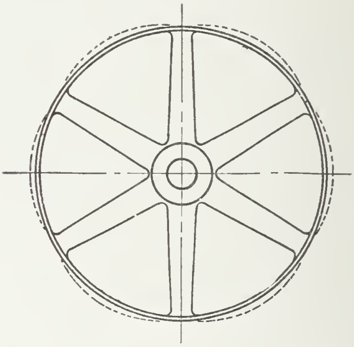


Fig. 20.

The tensile stress thus induced per square inch can be found by simple mechanics to be:

$$p = \frac{12wv^2}{g}; \qquad (17)$$

or, since $w = 0.26$ pound, and $g = 32.2$ feet per second,

$$p = 0.097\,v^2 \quad \left(\text{say } \frac{v^2}{10}\right); \qquad (18)$$

and, if p be taken equal to 1,000 pounds per square inch, which is as high as it is safe to work cast iron in this place,

$$v = 100 \text{ feet per second.} \qquad (19)$$

This shows the curious fact that the intensity of stress in the rim is directly proportional to the square of the linear velocity, and wholly independent of the area of cross-section. It is also to be noted that 100 feet per second is about the limit of speed for cast-iron pulleys to be safe against bursting.

If we wish to consider theoretically the rim together with the arms as actually connected to it, we get a much more complicated relation. This condition is shown in Fig. 20, where the rim, expanding more than the arms, bulges out between them. This makes the rim act something like a continuous beam uniformly loaded; but even then the resulting stress is not clearly defined on account of the variable stretch in the arms. Investigation on this basis is not needed further than to note that it is theoretically better, in the case of a split pulley, to make the joint close to the arms, rather than in the middle of a span.

Pulley Arms. The centrifugal force developed by the rim and arms tends to pull the arms from the hub. On the belt side, this is balanced to some extent by the belt wrap, which tends to compress the arm and relieve the tension. On the side away from the belt, the centrifugal action has full play, but the arm is usually of such cross-section that the intensity of this stress is very low. It may safely be neglected.

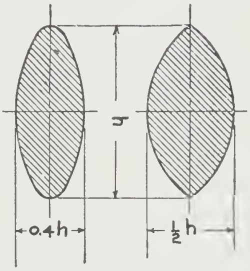


Fig. 21.

The rim being very thin in most cases, its distributing effect cannot be depended on, hence the driving force of the belt may be taken entirely by the arms immediately under the portion of the belt in contact with the pulley face. For a wrap of 180° this means that only one-half of the pulley arms can be considered as effective in transmitting the turning effort to the hub. Each of these arms is a lever fixed at one end to the hub and loaded at the other. A lever of this description is called a "cantilever" beam, its maximum moment existing at its fixed end. The load that each of these beams may be subjected to is $\frac{P}{\frac{N}{2}}$, and therefore the maximum external moment at the hub is $\frac{2PL}{N}$. From mechanics we know that the internal moment of resistance of any beam section is $\frac{SI}{c}$, and that equilibrium of the beam can be satisfied only when the external moment is equal to the internal moment of resistance of the beam section. Equating these two, we have:

$$\frac{2PL}{N} = \frac{SI}{c}. \qquad (20)$$

The arms of a pulley are usually of the elliptical or segmental cross-section, and may be of the proportions shown in Fig. 21. For either of these sections the fraction $\frac{I}{c}$ is approximately equal to $0.0393h^3$. For convenience (the error caused being on the safe side), L may be taken as equal to the full radius of the pulley R, whence

$$\frac{2PR}{N} = \frac{2(T_n - T_o)R}{N} = 0.0393Sh^3. \qquad (21)$$

in which S may be from 2,000 to 2,250 for cast iron

Taking moments about the center of the pulley, and solving for P_2, the force acting at the circumference of the hub, we have:

$$\frac{2PR}{N} = \frac{P_2D_1}{2};$$

or

$$P_2 = \frac{4PR}{ND_1} \qquad (22)$$

The area of an elliptical section is π times the product of the half axes. With the proportions of Fig. 21. this becomes:

$$a = \pi \times 0.2h \times 0.5h = \frac{\pi h^2}{10} \qquad (23)$$

Equating the external force to the internal shearing resistance, we have:

$$\frac{4PR}{ND_1} = \frac{\pi h^2 S_s}{10}$$

or,

$$S_s = \frac{40PR}{D_1N\pi h^2}, \qquad (24)$$

in which the shearing stress S_s may run from 1,500 to 1,800 for cast iron.

Although both bending and shearing stresses as calculated above exist at the base of the arms, the bending is, in practically every case, the controlling factor in the design of the arms. An arm-section large enough to resist bending would have a very low intensity of shear.

If the number of arms be increased indefinitely, we come to a continuous arm or web, in which the bending action is eliminated. It may still shear off at the hub, where the area of metal is the least, at minimum circumference. In this case the area under shearing stress is $\pi D_1 T$; and the force at the circumference of the hub, is

$$\frac{PR}{\frac{D_1}{2}} = \frac{2PR}{D_1}$$

Equating external force to internal shearing resistance, we have:

$$\frac{2PR}{D_1} = \pi D_1 T S_s;$$

or, $$S_s = \frac{2PR}{\pi D_1^2 T}. \qquad (25)$$

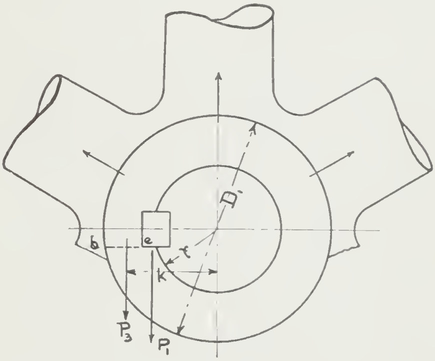


Fig. 22.

Pulley Hub. As in the case of the arms, centrifugal force does not play much part in the design of the hub of a pulley. The hub is designed principally to carry the key, and through it transmit the turning moment to the shaft. Considered thus, the hub may tear along the line of the key or crush in front of the key.

For example, in Fig. 22, if the connection with the lower arms be neglected, and the upper arms be held fast while a turning force P_1, at the surface of the shaft, is transmitted to the hub through the key, then the metal of the hub directly in front of the key is under crushing stress; and the metal along the line *eb*, from the corner to the outside, is under tensile stress. This condition is the worst that could possibly happen, because the bracing effect of the lower arms has been neglected, and the key is located between the arms.

Taking moments about the center of the shaft, the value of the force at the shaft circumference, or the "key pull," is:

$$P_1 = \frac{P\,R.}{r} \qquad \textbf{(26)}$$

Now $\frac{P_1}{P_3} = \frac{k}{r}$, k being the distance from the center of shaft to center of eb, and the area of metal which is subjected to the tearing action P_3 is $l \times eb$. Equating the external force to the internal resistance, and assuming that the stress is equally distributed over the area $l \times eb$, we have:

$$P_3 = \frac{r}{k} P_1 = \frac{r}{k} \times \frac{P\,R}{r} = S \times l \times e\,b;$$

or,

$$S = \frac{PR}{k \times l \times eb}. \qquad \textbf{(27)}$$

The intensity of crushing on the metal in front of the key, due to force P_1, depends upon the thickness of the key, and is properly discussed later under "Keys."

PRACTICAL MODIFICATION—Pulley Rim. The theoretical calculation for the thickness of the rim may give a thickness that could not be cast in the foundry, and the section in that case will have to be increased. As light a section as can be readily cast will usually be found abundantly strong for the forces it has to resist. A minimum thickness at the edge of the rim is about $\frac{3}{16}$ inch; and as the pulleys increase in size, the rim also must be made thicker; otherwise the rim will cool so much more quickly than the arms, that the latter, on cooling, will develop shrinkage cracks at the point of junction.

For a velocity of 6,000 feet per minute, we find from equation 18 that the tension in pounds per square inch, in the rim, due to centrifugal force, is 970. Though this in itself is a low value, yet the uncertain nature of cast iron, its condition of internal stress, due to casting, and the likely existence of hidden flaws and pockets, have established the usage of this figure as the highest safe limit for the peripheral speed of cast-iron pulleys. It is easily remembered that *cast-iron pulleys are safe for a linear velocity of about one mile per minute.*

To prevent the belt from running off the pulley, a "crown" or rounding surface is given the rim. A tapered face, which is more easily produced in the ordinary shop, may be used instead. This taper should be as little as possible, consistent with the belt staying on the pulley; $\frac{1}{2}$ inch per foot each way from the center is not too much for faces 4 inches wide and less; while above this width $\frac{1}{4}$ inch per foot is enough. As little as $\frac{1}{8}$ inch total crown has been found to be sufficient on a 24-inch face, but this is probably too little for general service.

Instead of being "crowned," the pulley may be flanged at the edges; but flanged pulley rims chafe and wear the edge of the belt.

The inside of the rim of a cast-iron pulley should have a taper of $\frac{1}{2}$ inch per foot to permit easy withdrawal from the foundry mould. This is known as "draft." If the pattern be of metal, or if the pulley be machine-moulded, the greater truth of the casting does not require that the inside of the rim be turned, as the pulley, at low speeds, will be in sufficiently good balance to run smoothly. For roughly moulded pulleys, and for use at high speeds, however, it is necessary that the rim be turned on the inside to give the pulley a running balance.

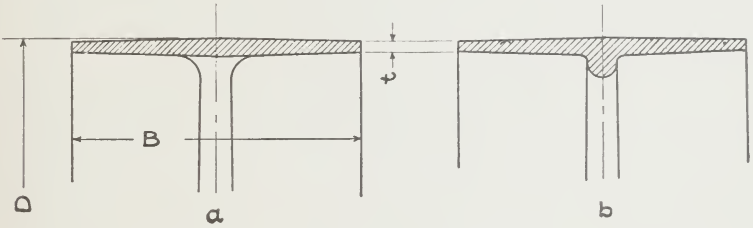


Fig. 23.

Fig. 23 shows a plain rim *a* also one stiffened by a rib *b*. Where heavy arms are used this rib is essential so that there will not be too sudden change of section at the junction of rim and arm, and consequent cracks or spongy metal.

Pulley Arms. The arms should be well filletted at both rim and hub, to render the flow of metal free and uniform in the mould. The general proportions of arms and connections to both hub and rim may perhaps be best developed by trial to scale on the drawing board. The base of the arm being determined, it may gradually taper to the rim, where it takes about the relation of $\frac{2}{3}$ to $\frac{3}{4}$ the dimensions chosen at the hub. The taper may be modified until it looks right, and then the sizes checked for strength.

Six arms are used in the great majority of pulleys. This number not only looks well, but is adapted to the standard three-jawed chucks and common clamping devices found in most shops. Elliptical arms look better than the segmental style. The flat, rectangular arm gives a very clumsy and heavy appearance, and is seldom found except on the very cheapest work.

A double set of arms may be used on an excessively wide face, but it complicates the casting to some extent.

Although a web pulley may be calculated for shear at the hub, yet it will usually be found that with a thickness of web intermediate between the thickness of the rim and that of the hub, which will satisfy the casting requirements, the requirements as to strength will be fully met.

Pulley Hub. The hub should have a taper of $\frac{1}{2}$ inch per foot draft, similar to that of the inside of the rim. The length of the hub is arbitrary, but should be ample to prevent rocking on the shaft. A common rule is to make it about $\frac{3}{4}$ the face width of the pulley.

The diameter of the hub, aside from the theoretical consideration given above, must be sufficient to take the wedging action of a taper key without splitting. This relation cannot well be calculated. Probably the best rule that exists is the familiar one that the hub should be twice the diameter of the shaft. This rule, however, cannot be literally adhered to, as it gives too small hubs for small shafts and too large ones for large shafts. It is always well to locate the key, if possible, underneath an arm instead of between the arms, thus gaining the additional strength due to the backing of the arm.

SPLIT PULLEYS.

ANALYSIS and THEORY. The split pulley is made in halves and provided with bolts through flanges and bosses on the hub for holding the two halves together. When the pulley is in place on the shaft, bolted up as one piece, it is subjected to the same forces as the simple pulley. Hence its general design follows the same principles, and we need only study the fastening of the two halves, and the effect of this fastening on the detail of rim and hub.

The simplest stress we have to consider on the rim bolts is one of pure tension, due to the centrifugal force of the halves of the pulley, A safe assumption to make is that the rim is free from the arms and hub, as in the simple pulley, and that the centrifugal force developed by it has to be taken by the rim bolts alone. In other words, consider the rim bolts as belonging entirely to the rim, and make them as strong as the rim, leaving the hub bolts to take the centrifugal force of the arms and hub, and the spreading tendency due to the key.

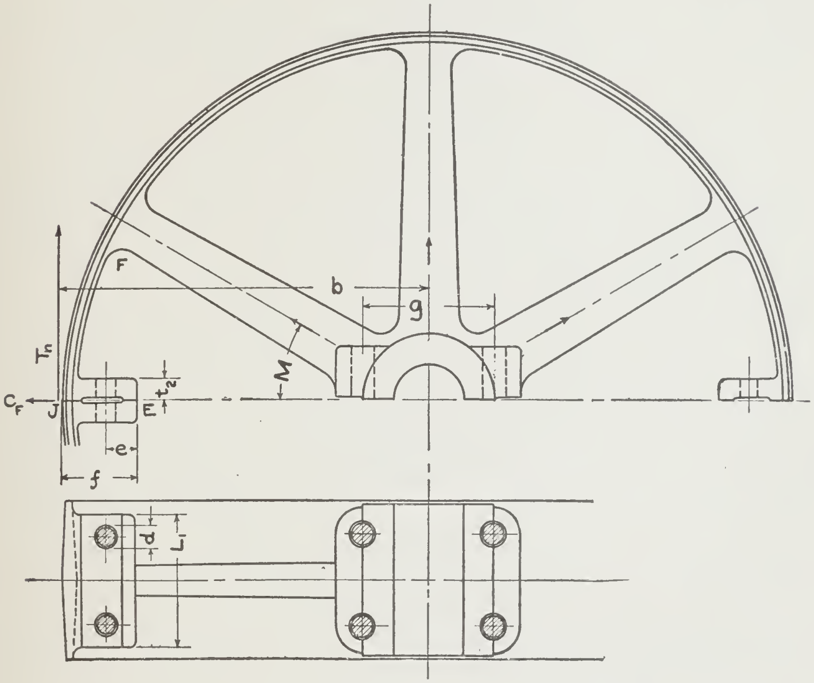


Fig. 24.

Another tensile stress is induced in the rim bolts by the fact, that, having made an open joint in the rim, and in addition placed the extra weight of lugs there, the centrifugal action at this point is increased, and at the same time a point of weakness in the rim introduced. Referring to Fig. 24, the rim flanges EJ tend to fly out due to the centrifugal force C_F. This tends to open the joint J at the outside of the rim ; to throw a bending stress on the rim, maximum at the point F ; and to "heel" the rim flanges about the point E. The rim bolts acting on the leverage e about the point E must resist these tendencies, and are thereby put in tension.

Referring to equation 18, we find the intensity of stress due to the centrifugal force of the rim in lbs. per square inch to be :

$$p = \frac{v^2}{10}.$$

If A is the sectional area of the rim in square inches, this means that the total strength of the rim is represented by $\frac{Av^2}{10}$. The strength of a bolt is represented by the expression $\frac{S\pi d_1^2}{4}$. If, now, there are n bolts in the flange, the total resisting force of the bolts is $\frac{nS\pi d_1^2}{4}$; and the equation representing equality of strength between rim and bolts is :

$$\frac{Av^2}{10} = \frac{nS\pi d_1^2}{4}, \quad \textbf{(28)}$$

from which, by a proper assumption of the fiber stress S, which should be low, the opening-up tendency of the joint being neglected, the diameter at the root of the thread d_1 may be calculated, and the nominal bolt diameter chosen. Reference to the table for strength of bolts, given in the chapter on Bolts, Studs, etc., will be found convenient.

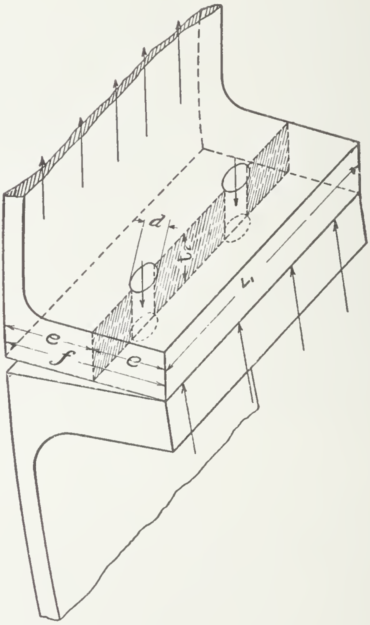


Fig. 25.

It is very doubtful if the tension on the flange bolts, due to the "heeling" about E can be calculated with sufficient accuracy to be of much value. It is probably better to assume S at a low value, say 4,000, and, in addition, for large and high-speed pulleys, stiffen the rim by running a rib between the flange and the adjacent arm. It is evident that if we make the rim so stiff that it cannot deflect, there will be no "heeling" about E; and the bolts will be well proportioned by the preceding calculation, giving them equal strength to that of the rim section.

For the bolt flange itself, any tendency to open at the joint J would cause it to act like a beam loaded at some point near its middle with the bolt load, and supported at J and E. This condition is shown in Fig. 25. Probably the weakest section would be along the line of the bolt centers. We have just noted that the carrying capacity of the bolts is $\frac{nS\pi d_1^2}{4}$. Hence, assuming that $e = \frac{1}{2}f$, which is about the worst case which could happen, we have a beam of length f loaded at the middle with $\frac{nS\pi d_1^2}{4}$ and supported at the ends. Equating the external moment to the internal moment, we have:

$$\frac{nS\pi d_1^2}{4} \times \frac{f}{4} = \frac{s(L_1 - nd)t_2^2}{6}. \qquad \textbf{(29)}$$

from which the fiber stress s in the flange may be calculated and judged for its allowable value.

L_1 may be assumed a little narrower than the pulley face; and t_2 from 1 inch to 2 inches or more, depending on the thickness of the rim.

The hub bolts doubtless assist the rim bolts in preventing the halves of the pulley from flying apart. They also clamp the hub tightly to the shaft, preventing any looseness on the key. Their function is a rather general one; and the specific stress which they receive is practically impossible to calculate. As a matter of fact, if the hub bolts were left out entirely, the pulley would still drive fairly well, but general rigidity and steadiness would be impaired. Hence the size of the hub bolts is more a practical question than one involving calculation. The rim bolts should be figured first, and their size determined on; then the hub bolts can be judged in proportion to the rim bolts, the diameter of shaft, the thickness and length of the hub, and the general form of the pulley. Often appearance is the deciding factor, it being manifestly inconsistent to associate small fastenings with large shafts or hubs, even though the load be actually small.

PRACTICAL MODIFICATION. Practical considerations are chiefly responsible for the location of the joint in a split pulley between the arms instead of directly at the end of an arm, where theoretically it would seem to be required. It is usually more convenient in the foundry and machine shop to have the joint between the arms; so we generally find it placed there, and strength provided to permit this. It is possible, however, to provide a double arm, or a single split arm, in which case the joint of the pulley comes at the arm, and the "heeling" action of the rim flanges is prevented.

The rim bolts should be crowded as close as possible to the rim in order to reduce the stress on them, and also the stress in the flange itself. The practical point must not be forgotten, however that the bolts must have sufficient clearance to be put into place beneath the rim.

While it is evident that the rim bolts are most effective in taking care of the centrifugal action of the halves, yet in small split pulleys it is quite common to omit the rim bolts and to use the hub bolts for the double purpose of clamping the shaft and holding the two halves together. The pulley is cast with its rim continuous throughout the full circle, and it is machined in this form. It is then cracked in two by a well-directed blow of a cold chisel, the casting being especially arranged for this along the division line by cores so set that but a narrow fin of metal holds the two parts together. This provides sufficient strength for casting and turning, but permits the cold chisel to break the connection easily.

SPECIAL FORMS OF PULLEYS.

The plain cast-iron pulley has been used in the foregoing discussion as a basis of design. A pulley is, however, such a common commercial article, and finds such universal use, that special forms, which can be bought in the open market, are not only cheaper but better than the plain cast-iron pulley, at least for regular line-shaft work.

Cast iron is a treacherous and uncertain material for rims of pulleys. It is not well suited to high fiber stresses; hence the range of speed permissible for pulley rims of cast iron is limited. Steel and wrought iron, having several times the tensional strength of cast iron, and being, moreover, much more nearly homogeneous in texture, are well suited for this work; one of the best pulleys on the market consists of a steel rim riveted to a cast-iron spider. Such an arrangement combines strength and lightness, without increasing complication or expense.

The all-steel pulley is a step further in this direction. Here the rim, arms, and hub are each pressed into shape by specially devised machinery, then riveted and bolted together. This pulley is strictly a manufactured article, which could not compete with the simpler forms unless built in large quantities, enabling automatic machinery to be used. Large numbers of pulleys are built in this way, and are put on the market at reasonable prices.

Wood-rim pulleys have been made for many years, and, except for their clumsy appearance, are excellent in many respects. The rim is built up of segments in much the same way as an ordinary pattern is made, the segments being so arranged that they will not shrink or twist out of shape from moisture. The hubs may be of cast iron, bolted to wooden webs, and carrying hardwood split bushings, which may be varied in bore within certain limits so as to fit different sizes of shafting. The wooden pulley is readily and most often used in the split form, thus enabling it to be put in position easily at any point of a crowded shaft. It is often merely clamped in place, thus avoiding the use of keys or set screws, and not burring or roughening the shaft in any way.

PROBLEMS ON PULLEYS.

1. Calculate the tensile stress due to centrifugal force in the rim of a cast-iron pulley 30 inches in diameter, at 500 revolutions per minute.

2. The driving force of a belt on a 36-inch pulley is 800 lbs., and the belt wrap about 180°. Calculate proportions of el-

liptical arms to resist bending, the allowable fiber stress being 2,000.

3. A pulley 12 inches in diameter, $\frac{5}{8}$-inch web, 4-inch diameter hub, transmits 25 horse-power at a belt speed of 3,000 ft. per minute. Calculate the maximum shearing stress in the web.

4. In Fig. 24 assume the following data: $L_1 = 7$ inches; $t_2 = 1$ inch; $e = 1\frac{1}{2}$ inches; $f = 3$ inches; area of rim = 3 sq. in.; allowable tensile stress in rim 1,000 lbs. per sq. in. Calculate the diameter of the rim bolts.

5. Calculate the fiber stress in the rim bolt flange along the line of the bolts.

SHAFTS.

NOTATION—The following notation is used throughout the chapter on Shafts:

A_0 = Angular deflection (degrees).
B = Simple bending moment (inch-lbs.).
B_e = Equivalent bending moment (inch-lbs.).
c = Distance from neutral axis to outer fiber (inches).
d, d_0, d_2, d_3, d_4 = Diameters of shaft (inches).
d_1 = Internal diameter of shaft (inches).
E = Direct modulus of elasticity (a ratio).
e = Transverse deflection (inches).
G = Transverse modulus of elasticity (a ratio).
H = Horse-power (33,000 ft.-lbs. per minute).
I = Moment of inertia.
K = Distance between bearings (inches).
L = Length along shaft (inches).
L_1, L_2 = Length of bearings (inches.)
M = Distance between bearings (feet.)
N = Number of revolutions per minute.
P = Driving force of belt (lbs.).
P_1 = Load applied as stated (lbs.).
R = Radius at which load as stated acts (inches).
S = Fiber stress, tension, compression, or shearing (lbs. per sq. in.).
T = Simple twisting moment (inch-lbs.).
T_e = Equivalent twisting moment (inch-lbs.).
T_n = Tension in tight side of belt (lbs.).
T_o = Tension in loose side of belt (lbs.).
W = Load applied as stated (lbs.).

ANALYSIS. The simplest case of shaft loading is shown in Fig. 26. The equal forces W, similarly applied to the disc at the distance R from its center, tend to twist the shaft off, the tendency being equal at all points of the length L between the disc and the post, to which the shaft is rigidly fastened. The fastening to the post, of course, in this ideal case, takes the place of a resisting member of a machine. A state of pure torsion is induced in the shaft; and any element, such as *ca*, is distorted to the position *cb*, *aob* being the angular deflection for the distance L.

The case of Fig. 27 is illustrative of what occurs when a belt pulley is substituted for the simple disc. Here the twisting action is caused by the driving force of the belt, which is $T_n - T_o = P$, acting at the radius R. Torsion and angular deflection exist in the shaft, as in Fig. 26. In addition, however, another stress of a different kind has been introduced; for not only does the shaft tend to be twisted off, but the forces T_n and T_o, acting together, tend to bend the shaft, the bending moment varying with every section of the shaft, being nothing at the point *o*, and maximum at the point *c*. This combined action is the most common of any that we find in ordinary machinery, occurring in nearly every case with which we have to deal.

In Fig. 27, if the forces T_n and T_o be made equal, there will be no tendency at all to twist off the shaft, but the bending will remain, being maximum at the point *c*. This condition is illustrative of the case of all ordinary pins and studs in machines. In this sense, a pin or a stud is simply a shaft which is fixed to the frame of the machine, there being no tendency to turning of the pin or stud itself. The same condition would be realized if the disc in Fig. 27 were loose upon the shaft. In that case, the bending moment would be caused by $T_n + T_o$ acting with the leverage L. Of course there would have to be some resistance for $T_n - T_o$ to work against, in order that torsion should not be transmitted through the shaft. This condition might be introduced by having a similar disc lock with the first one by means of lugs on its face, thus receiving and transmitting the torsion.

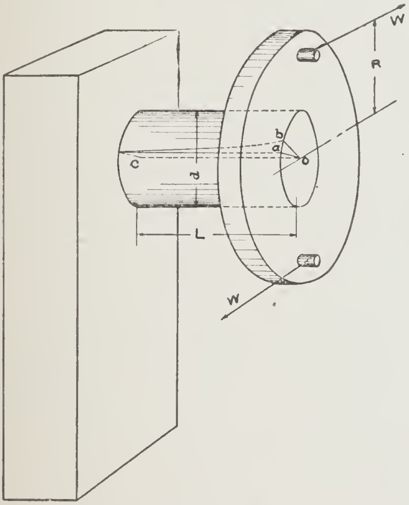


Fig. 26.

If the distance L becomes very great, both the angular deflection due to twisting, and the sidewise deflection due to bending, become excessive, and not permissible in good design. This trouble is remedied by placing a bearing at some point closer to the disc, which, as it decreases L, of course, decreases the bending moment and therefore the transverse deflection. The angular deflection can be decreased only by bringing the resistance and load nearer together.

The above implies, of course, that the diameter of the shaft is not changed, it being obvious that increase of diameter means increase of strength and corresponding decrease of both angular and transverse deflection.

If the speed of the shaft be very high, and the distance between bearings, represented by L, be very great, the shaft will take a shape like a bow string when it is vibrated, and smooth action cannot be maintained.

It is necessary to carry the cases of Figs. 26 and 27 but a single step farther to illustrate the actual working conditions of shafting in machines. Suppose the rigid post to have the shaft passing clear through it, and to act as a bearing, so that the shaft can freely rotate in it, the resistance being exerted somewhere beyond. The twisting moment will be unchanged, also the bending moment; but the effect of the bending moment will be on each particle of the shaft in succession, now putting compression on a given particle, and then tension, then compression again, and so on, a complete cycle being performed for each revolution. This

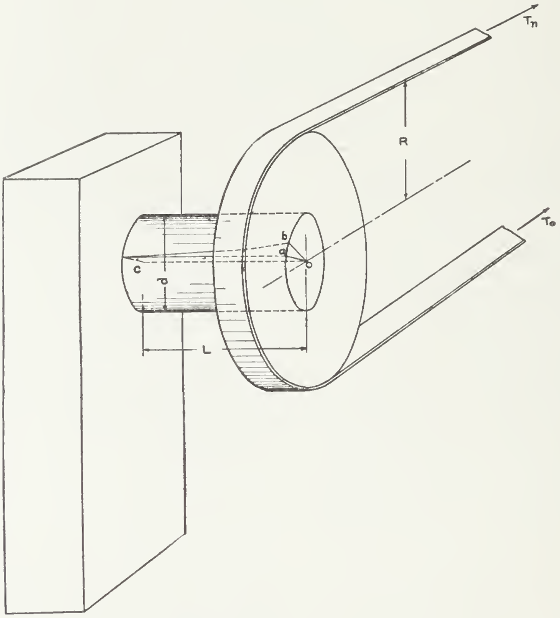


Fig. 27.

brings out a very important difference between the bending stress in pins and the bending stress in rotating shafts. In the one case the bending stress is non-reversing; in the other, reversing; and a much higher fiber stress is permissible in the former than in the latter.

THEORY—Simple Torsion. In the case of simple torsion the stress induced in the shaft is a shearing one. The external moment acts about the axis of the shaft, or is a polar moment; hence in the expression for the moment of the internal forces, the polar moment of inertia must be used. Now, from mechanics we have:

$$T = \frac{S\,I}{c};$$

and $$\frac{I}{c} = \frac{d^3}{5.1} \text{ (for circular section of diameter } d\text{);}$$

therefore, $$T = \frac{Sd^3}{5.1}, \qquad (30)$$

from which the diameter for any given twisting moment and fiber stress can readily be found.

For a hollow shaft this expression becomes:

$$T = \frac{S(d_o^4 - d_1^4)}{5.1 d_o}. \qquad (31)$$

Simple Bending. The stresses induced in a pin or shaft under simple bending are compression and tension. The external moment in this case is transverse, or about an axis across the shaft; hence the direct moment of inertia is applicable to the equation of forces.

$$B = \frac{S\,I}{c};$$

and $$\frac{I}{c} = \frac{d^3}{10.2} \text{ (for circular section of diameter } d\text{);}$$

therefore, $$B = \frac{Sd^3}{10.2}. \qquad (32)$$

For a hollow shaft or pin this expression becomes:

$$B = \frac{S(d_o^4 - d_1^4)}{10.2 d_o}. \qquad (33)$$

Combined Stresses. In the greater number of cases met with in practice, we find two or more simple stresses acting at the same time, and, although the shaft may be strong enough for any one of them alone, it may fail under their combined action. The most common cases are discussed below.

Tension or Pressure Combined with Bending. In Fig. 28, the load W produces a tension acting over the whole area of *d*, due to its direct pull. It also produces a bending action due to the leverage R, which puts the fibers at B in tension and those at the opposite side in compression. It is evident, therefore, that by taking the algebraic sum of the stresses at either side we shall obtain the net stress. It is also evident that the greatest and controlling stress will occur on the side where the stresses add, or on the tension side. Hence, from mechanics,

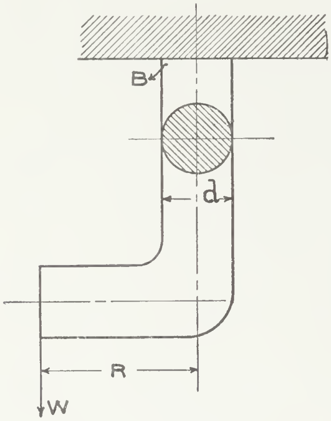


Fig. 28.

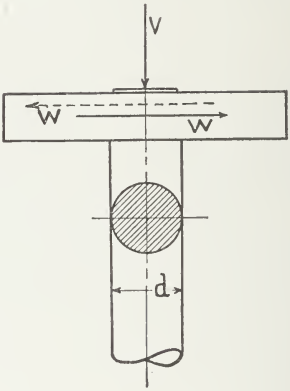


Fig. 29

$$W = \frac{\pi d^2 S}{4};$$

or, $$S = \frac{4W}{\pi d^2} \text{ (due to direct tension).} \qquad \textbf{(34)}$$

Also, $$WR = \frac{Sd^3}{10.2};$$

or, $$S = \frac{10.2\ WR}{d^3} \text{ (due to bending).} \qquad \textbf{(35)}$$

Hence the combined tensional stress acting at the point B, or, in fact, at any point on the extreme outside of the vertical shaft toward the force W, is:

$$S = \frac{4W}{\pi d^2} + \frac{10.2\ WR}{d^3}. \qquad (36)$$

If W acted in the opposite direction, the greatest stress would still be at the side B, but would be a compression instead of a tension, of the same magnitude as before.

Tension or Compression Combined with Torsion. In Fig. 29, V might be the end load on a vertical shaft; and the two forces W might act in conjunction with it as in the case of Fig. 26, at the radius R. This case is not very often met with. It is usually possible to combine the moments, find an equivalent moment of a simple kind, and use the corresponding simple fiber stress. In the case in question we have a direct stress to be combined with a shearing stress, and mechanics gives us the following solution:

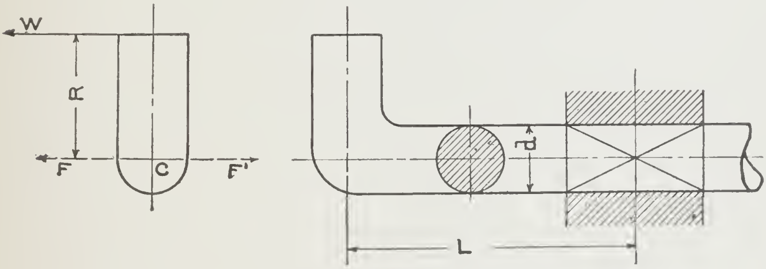


Fig. 30.

Let S_s = simple shearing stress (lbs. per sq in.).
Let S_c = simple compressive stress (lbs. per sq. in.).
Let S_{rs} = resultant shearing stress (lbs. per sq. in.).
Let S_{rc} = resultant compressive stress (lbs. per sq. in.).

We then have :

$$2WR = \frac{S_s d^3}{5.1};$$

or,

$$S_s = \frac{5.1(2WR)}{d^3}. \qquad (37)$$

Also,

$$V = \frac{\pi d^2 S_c}{4};$$

or,

$$S_c = \frac{4}{\pi d^2} V. \qquad (38)$$

Now, from a solution given in simplest form in "Merriman's Mechanics"—which the student may consult, if desired—values for the resultant stresses can be found. Whichever of these is the critical one for the material used, should form the basis for its diameter:

$$S_{rs} = \sqrt{S_s^2 + \frac{S_c^2}{4}}. \qquad (39)$$

Also,

$$S_{rc} = \frac{S_c}{2} + \sqrt{S_s^2 + \frac{S_c^2}{4}}. \qquad (40)$$

Bending Combined with Torsion. In Fig. 30, the load W acts not only to twist the shaft off, but also presses it sidewise against the bearing. As it is usually customary to figure the maximum moment as taking place at the center of the bearing, the length L, which determines the bending moment, is taken to that point. The theory of the stress induced in this case is complicated. In order to make the magnitude of the moments clearer, let us introduce the two equal and opposite forces F and F^1, each equal to W, at the point C. We can evidently do this without changing the equilibrium of the shaft in any way. We now see that W and F^1 act as a couple giving a twisting moment WR; and that F acts with a leverage L, producing a bending moment FL = WL, at the middle of the bearing.

If, now, we find an equivalent twisting moment, or an equivalent bending moment, which would produce the same effect on the fibers of the shaft as the two combined, we can treat the calculation of the diameter as a simple case, and proceed as in the cases of simple torsion and simple bending considered above. This relation is given us in mechanics:

$$B_e = \frac{B}{2} + \frac{1}{2}\sqrt{B^2 + T^2}. \qquad (41)$$

$$T_e = B + \sqrt{B^2 + T^2}. \qquad (42)$$

These expressions are true in relation to each other, on the assumption that the allowable fiber stress S is the same for tension, compression, and shearing. For the material of which shafts are usually made, this is near enough to the truth to give safe and practical results. Using the expressions for internal moments of resistance as previously noted for circular sections, we then have :

$$B_e = \frac{Sd^3}{10.2}. \qquad (43)$$

Also,

$$T_e = \frac{Sd^3}{5.1}. \qquad (44)$$

Either equation may be used ; the diameter d will result the same whichever equation is taken. For the sake of simplicity, equation 42 is generally preferred, equation 44 being taken in conjunction with it.

The expression $\sqrt{B^2 + T^2}$ is one that would be a long and tedious task to calculate. By inspection it is readily seen that this quantity can be graphically represented by means of a right-angled triangle having B and T as the sides. We may then lay down on a piece of paper, to some convenient scale, the moments B and T as the sides of a right-angled triangle, when, upon measuring the hypothenuse, we can easily read off to the same scale $\sqrt{B^2 + T^2}$. Even if the drawing is made to a small scale, the accuracy of the reading will be sufficient to enable the value for d to be solved very closely. This graphical method is illustrated in Part I.

Deflection. For a shaft subjected to pure torsion, as in Fig. 26, the angular deflection due to the load may be carried to a certain point before the limit of working fiber stress is exceeded. The equation worked out from mechanics for this condition, is:

$$A^\circ = \frac{584\ TL}{Gd^4}, \qquad (45)$$

which at once gives the number of degrees of angular deflection for a shaft whose modulus of elasticity, torsional moment, and length are known.

The shearing modulus of elasticity of ordinary shaft steel runs from 10,000,000 to 13,000,000, giving as an average about 12,000,000.

By the well-known relation of "Hooke's law" (stresses proportional to strains within the elastic limit of the material), we have:

$$\frac{A}{360^\circ} = \frac{SL}{\pi G d};$$

or

$$S = \frac{A \pi G d}{360\ L}. \qquad (46)$$

A twist of one degree in a length of twenty diameters is a usual allowance. Substituting A = 1, L = 20d, and G = 12,000,000, we have:

$$S = 5{,}240 \text{ (nearly)}. \qquad (47)$$

This is a safe value for shearing fiber stress in steel. In fact, in calculations for strength, even for reversing stresses, the usual figure is 8,000 (lbs. per square inch), thus indicating that the relation of one degree to twenty diameters is well within the limit of strength.

For a hollow shaft the above formula becomes :

$$A^\circ = \frac{584\ TL}{G(d_0^4 - d_1^4)}. \qquad (48)$$

Transverse deflection occurs when the shaft is subjected to a bending moment. It may therefore exist alone or in conjunction with angular deflection. Transverse deflection of shafts, however, rarely exists up to the point of limiting fiber stress, because before that point is reached the alignment of the shaft is so disturbed that it is not practicable as a device for transmitting power. A transverse deflection of .01 inch per foot of length is a common allowance ; but it is impossible to fix any general limit, as in many cases this figure, if exceeded, would do no harm, while in others—such as heavily loaded or high-speed bearings—even the figure given might be fatal to good operation.

The formula for transverse deflection, deduced from mechanics, varies with the system of loading. The three most common conditions only are given below, reference to the handbook being necessary if other conditions must be satisfied:

Fixed at one end, loaded at the other,

$$e = \frac{WL^3}{3\ EI}. \qquad (49)$$

Supported at ends, loaded in middle,

$$e = \frac{WL^3}{48\,EI}. \tag{50}$$

Supported at ends, loaded uniformly,

$$e = \frac{5WL^3}{384\,EI}. \tag{51}$$

For transverse deflection the direct modulus of elasticity must be used, for the fibers are stretched or compressed, instead of being subjected to a shearing action. The most usual value of the direct modulus of elasticity for ordinary steel is 30,000,000, and is denoted in most books by the symbol E. Both the shearing and direct moduli of elasticity are really nothing but the ratio of the stress to the strain produced by that stress, it being assumed that the given material is perfectly elastic. A material is supposed to be perfectly elastic up to a certain limit of stress, and it is within this limit that the relation as above holds good.

Expressed in the form of an equation this would be :

$$E = \frac{S}{\frac{e}{L}} = \frac{SL}{e}. \tag{52}$$

Centrifugal Whirling. If a line shaft deflect but slightly, due to its own weight, or the weight or pressure of other bodies upon it, and then be run at a high speed, the centrifugal force set up increases the deflection, and the shaft whirls about the geometrical line through the centers of the bearings, causing vibration and wear in the adjoining members. It is evident that the practical remedy for this tendency in a shaft of given diameter and speed is to locate the bearings sufficiently close to render the action of small effect.

Many formulæ might be given for this relation, each being based on different assumptions. Perhaps as widely applied and as simple as any, is the "Rankine" formula, which sets the limit of length between bearings for shafts not greatly loaded by intermediate pulleys or side strains :

$$M = 175 \sqrt{\frac{d}{N}}. \tag{53}$$

Horse-Power of Shafting. Horse-power is a certain specific rate of doing work, *viz.*, 33,000 foot-pounds per minute. Hence, to find the horse-power that a shaft will transmit, we must first find the work done, and then relate it to the speed. Take, for example, the case of a pulley, the symbols being the same as before—namely, P = driving force at rim of pulley (lbs.); R = radius of pulley (inches); N = number of revolutions per minute; and H = horse-power. Then,

$$\text{Work} = \text{force} \times \text{distance} = P \times (2\,\pi\,RN) = H \times 33{,}000 \times 12;$$

or,

$$PR = \frac{63{,}025H}{N}. \qquad \textbf{(54)}$$

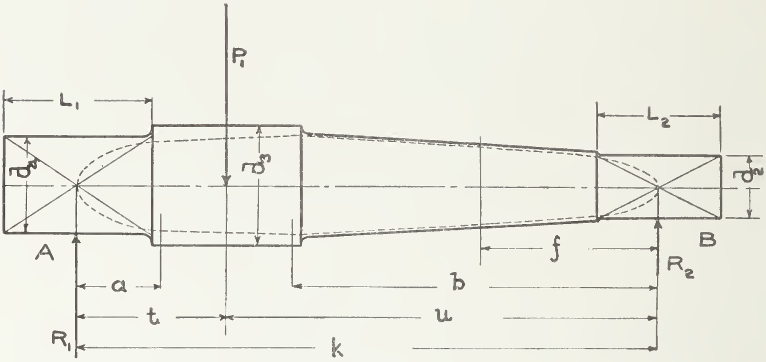


Fig. 31.

This is one of the most useful equations for calculations involving horse-power. By it the number of inch-pounds torsion for any horse-power can be at once ascertained.

It should be clearly noted, however, that in this equation the bending moment does not enter at all. Hence any shaft based in size on *horse-power alone*, is based on *torsional moment alone*, bending moment being entirely neglected. In many cases the bending moment is the controlling one as to limiting fiber stress. Hence empirical shafting formulæ depending upon the horse-power relation are unsafe, unless it is definitely known just what torsional and bending moments have been assumed.

The only safe way to figure the size of a shaft is to find accurately what torsional moment and bending moment it has to sustain, and then combine them according to equation 41 or 42 introducing the element of speed as basis for assumption of a high or low working fiber stress.

PRACTICAL MODIFICATION. The practical methods of handling the theoretical shaft equations have reference to the fit of the shaft within the several pieces upon it. The running fit of a shaft in a bearing is usually considered to be so loose that the shaft could freely deflect to the center of the bearing. This is doubtless an extreme view of the case, but it is the only safe assumption. Hence a shaft running in bearings (see Fig. 31) is supposed to be supported at the centers of those bearings, and its theoretical strength is based on this supposition.

For a tight or driving fit upon the shaft, a safe assumption to make is that there is looseness enough at the ends of the fit to permit the shaft to be stressed by the load a short distance within the faces of the hub, say from $\frac{1}{2}$ inch to 1 inch. For example, referring to Fig. 31, suppose P_1 to be the transverse load, exerted through a hub fast upon the part of the shaft d_3. Taking moments about the center of one bearing, and solving for the reaction at the center of the other, we have:

$$P_1\,u = R_1\,K;$$

or,

$$R_1 = \frac{P\ u}{K}. \qquad (55)$$

Also,

$$P_1\,t = R_2\,K;$$

or,

$$R_2 = \frac{P_1\,t}{K}. \qquad (56)$$

Now, as far as the part of shaft d_3 is concerned, it may depend for its size on the bending moment $R_2\,b$, or on $R_1\,a$. The reason the lever arm is not taken to the point directly under the load P_1, is because it is not practically possible to break the shaft at that point, on account of the reinforcement of the hub, which is tightly fitted upon it. Trying these moments to see which is the greater, we shall find that the greater moment always occurs in connection with the longer lever arm. Hence $R_2\,b$ will be greater than $R_1\,a$. We then write the equation of *external moment = internal moment:*

$$R_2\,b = \frac{S\,d_3^{\,3}}{10.2};$$

or,
$$d_3 = \sqrt[3]{\frac{10.2\ R_2 b}{S}}. \qquad \textbf{(57)}$$

For the size of bearing A we have the maximum bending moment:

$$R_1 \frac{L_1}{2} = \frac{S\, d_4^3}{10.2};$$

or,
$$d_4 = \sqrt[3]{\frac{10.2\ R_1 L_1}{2\ S}}. \qquad \textbf{(58)}$$

For the size of bearing B we have the maximum moment:

$$R_2 \frac{L_2}{2} = \frac{S\, d_2^3}{10.2};$$

or,
$$d_2 = \sqrt[3]{\frac{10.2\ R_2 L_2}{2\ S}}. \qquad \textbf{(59)}$$

The above calculations are, of course, on the assumption that no torsion is transmitted either way through this axle. We should in that case have combined torsion and bending. This has been made sufficiently clear in preceding paragraphs and in Part I, to require no further illustration.

The dotted line in Fig. 31 shows the theoretical shape the axle should take under the assumed conditions. The practical modification of this shape is obvious. At the shoulders of the shaft the corners should not be sharp, but carefully filleted, to avoid the possible starting of a crack at those points.

Often the diameter of certain parts of a shaft may be larger than strength actually calls for. For example, in Fig. 31, the part d_3 need only be as large as the dotted line; but it is obvious that unless the key is sunk in the body of the shaft, the hub could not be slipped into place over the part d_4. If, however, the diameter d_3 be made large enough so that the bottom of the key will clear d_4, the rotary cutter which forms the key way in d_3 will also clear d_4, and the key way can be more easily produced.

In cases where fits are not required to be snug, a straight shaft of cold-rolled steel is commonly used. Here any parts fastened on the middle of the shaft have to be driven over a considerable length of the shaft before they reach their final position. Moreover, there is no definite shoulder to stop against, and measurement has to be resorted to in locating them.

It does not pay to turn any portion of a cold-rolled shaft, unless it be the very ends, for relieving the "skin tension" in such material is sure to throw the shaft out of line and necessitate subsequent straightening.

Turned-steel shafts for machines may with advantage be slightly varied in diameter wherever the fit changes; and although the production of shoulders costs something, yet it assists greatly in bringing the parts to their exact location, and enables the workman to concentrate his best skill on the fine bearing fits, and to save time by rough-turning the parts that have no fits.

Hollow shafts are practicable only for large sizes. The advantages of removing the inner core of metal, aside from some specific requirement of the machine, are that it eliminates all possibility of cracks starting from the checks that may exist at the center, permits inspection of the material of a shaft, and, in case of hollow-forged shafts, gives an opening for the forging mandrel. In the last case, the material is improved by a rolling process.

The material most common for use in machine shafting is the ordinary "Machinery Steel," made by the Bessemer process. This steel is apt to be "seamy," and often contains checks and flaws that are detected only upon sudden and unexpected breakage of a part apparently sound. This characteristic is a result of the process employed in the manufacture of the steel, and thus far has never been wholly eliminated. **Bessemer steel** is, nevertheless, a very useful material, and the above weakness is not so serious but that this kind of steel can be used with success in the great majority of cases.

When a more homogeneous shaft is desired, **open-hearth steel** is available. This is a more reliable material to use than the Bessemer, and costs somewhat more. It makes a stiff, true, fine-surfaced shaft, high-grade in every respect. It is usually specified for armature shafts of dynamos and motors.

Steels of special strength, toughness, and elasticity are made under numerous processes. **Nickel steel** is perhaps the most conspicuous example. While for this steel a high price has to be paid, yet its great strength, in connection with other valuable qualities, makes it a material extremely valuable for service where light weight is essential, or where contracted space demands small size.

The range of strength of these various steels is so great that it is well-nigh useless to go into a discussion of it here. Reference should be had to the extended discussions of the handbooks, and to special trade pamphlets. A study of the possibilities of steel in its various forms for use in shafting, is very valuable as a basis for design, as it can almost be said that a machine consists chiefly of a "collection of shafts with a structure built round them." The shafts are like a core, and evidently the size of the core determines the shell about it.

PROBLEMS ON SHAFTS.

1. Required the twisting moment on a shaft that transmits 30 horse-power at 120 revolutions per minute.

2. Find the diameter of a steel shaft designed to transmit 50 horse-power at 150 revolutions per minute.

3. Assuming same data as in Problem 1, find the diameters of a hollow shaft for a value of $S = 8,000$.

4. A belt on an idler pulley embraces an angle of 120 degrees. Assuming tension of belt 1,000 pounds on each side, and pulley located midway between bearings, which are 30 inches from center to center, what is the diameter of shaft required?

5. Calculate the diameter of a steel shaft designed to transmit a twisting moment of 400,000 inch-pounds and also to take a bending moment of 300,000 inch-pounds.

6. Find the angular deflection in a 4-inch shaft 20 feet long when subjected to a load of 5,500 pounds applied to an arm of 30-inch radius. Assume transverse modulus of elasticity equal to 12,000,000.

7. The overhung crank of a steam engine has a force of 32,000 lbs. at the center of the crank pin, which is 12 inches from the center of the shaft bearing, measured parallel to the shaft. The radius of crank arm is 10 inches. Assume S equal to 10,000. Calculate the diameter of the crank shaft.

8. On a short, vertical steel shaft the load is 5,000 pounds. A gear, 36 teeth, $1\frac{1}{2}$ diametral pitch, at top of shaft, transmits a load of 4,000 pounds at the pitch line. Safe shear = 7,500. What is the diameter of the shaft?

SPUR GEARS.

NOTATION—The following notation is used throughout the chapter on Spur Gears:

b = Breadth of rectangular section of arm (inches).

M, M_1 = Revolutions per minute.

μ = Coefficient of friction between teeth.

C = Width of arm extended to pitch line (inches).

c = Distance from neutral axis to outer fiber (inches).

D = Pitch diameter of gear (inches).

F = Face of gear (inches).

f = Clearance of tooth at bottom (inches).

G = Thickness of arm extended to pitch line (inches).

H = Thickness of tooth at any section (inches).

h = Depth of rectangular section of arm (inches).

I = Moment of inertia.

K = Thickness of rim (inches).

L = Distance from top of tooth to any section (inches).

N = Number of teeth.

n = Number of arms.

P = Diametral pitch (teeth per inch of diameter).

P^1 = Circular pitch (inches).

Q, Q_1 = Normal pressure between teeth (lbs.).

R, R_1 = Resultant pressure between teeth (lbs.).

r, r_1 = Radius of pitch circles (inches).

S = Fiber stress of material (lbs. per sq. in.).

s = Addendum of tooth (inches) = Dedendum of tooth.

t = Thickness of tooth at pitch line (inches).

W = Load at pitch line (lbs.).

y = Coefficient for "Lewis" formula.

ANALYSIS. If a cylinder be placed on a plane surface, with its axis parallel to the plane, an attempt to rotate the cylinder about its axis would cause it to roll on the plane.

Again, if two cylinders be provided with axial bearings, and be slightly pressed together, motion of one about its axis will cause a similar motion of the other, the two surfaces rolling one on the other at their common tangent line. If moved with care, there will be no slipping in either of the above cases—which is explained by the fact that no matter how smooth the surfaces may appear to be, there is still sufficient roughness to make the little irregularities interlock and act like minute teeth.

The magnitude of the force possible to be transmitted depends not only on the roughness of the surfaces, but on the amount of pressure between them. Suppose that one cylinder is a part of a hoisting drum, on which is wound a rope with a weight attached. We can readily make the weight so great that, no matter how hard we press the two cylinders together, the driving cylinder will not turn the hoisting cylinder, but will slip past it. If now, instead of increasing the pressure, which is detrimental both to cylinders and bearings of same, we increase the coarseness of the surfaces, or, in other words, put teeth of appreciable size on these surfaces, we attain the desired result of positively driving without excessive side pressure.

These artificial projections, or teeth, must fit into one another; hence the surfaces of the original cylinders, having been broken up into alternate projections and hollows, have entirely disappeared to the eye; they nevertheless exist as ideal or imaginary surfaces, which roll together with the same surface velocities as if in bodily form, provided that the curves of the teeth are correctly formed. Several mathematical curves are available for use as tooth outlines, but in practice the **involute** and **cycloidal curves** are the only ones used for this purpose.

The ideal surfaces are known as **pitch cylinders** or **pitch circles.** In Fig. 32 is shown an end view of such a pair of cylinders in contact at their pitch point P. In gear calculations we assume that there is no slip between the pitch circles, acting as driving cylinders; hence the speeds of the two pitch circles at the pitch point are equal. If M and M_1 be the revolutions per minute of the cylinders respectively, r and r_1 their radii, then

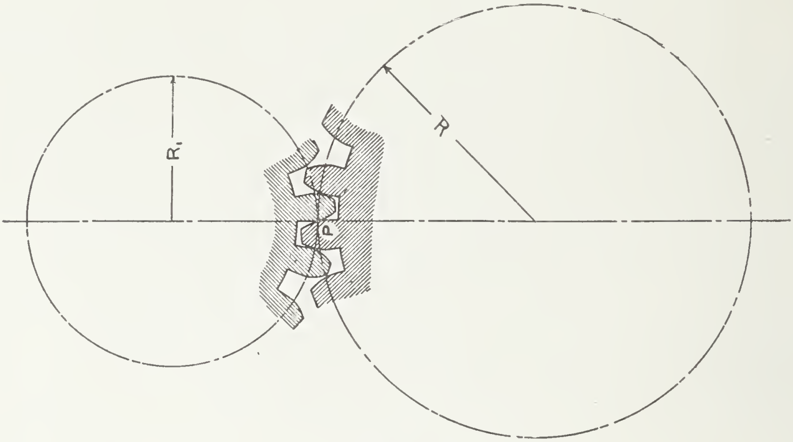


Fig. 32.

$$2\pi r M = 2\pi r_1 M_1;$$

or,
$$\frac{M}{M_1} = \frac{r_1}{r}. \qquad (60)$$

That is, the number of revolutions varies inversely as the radii.

The simple calculation as above is the key to all calculations involving gear trains in reference to their speed ratio.

Fig. 33 represents cycloidal teeth in the two extreme positions of beginning and ending contact. The normal pressure Q or Q_1 between the teeth in each position acts through the pitch point O, as it must always do in order to insure the condition of ideal rolling of the pitch circles, and the velocity ratio proportional to $\frac{r'_1}{r'}$ As the surfaces of the teeth slide together, frictional resistance is produced at their point of contact. This force is widely variable, depending on the material and condition of the tooth surfaces, whether smooth and well lubricated, or rough and gritty. As this resistance acts in conjunction with the normal force between the teeth, we may construct a parallelogram of forces on these two as a base, the resultant pressure between the teeth being slightly changed thereby, as shown in Fig 33.

Assuming a coefficient of friction μ, the force of friction is μ Q or μ Q_1 and the resultant pressure R or R_1.

Tooth B of the FOLLOWER is therefore under a heavy bending moment measured by the product RL, L being the perpendicular distance from the center of the tooth at its base to the line of the force. This tooth also has a relatively small compressive stress due to the resolved part of R along the radius, and a relatively small shearing stress due to the resolved part of R along a tangent to the pitch circle.

Tooth D of the driven wheel or FOLLOWER has a relatively large shearing stress, a small bending moment, and practically no direct compressive stress.

Tooth A of the driving wheel or DRIVER has a relatively large shearing stress, a small bending moment, and small compressive stress.

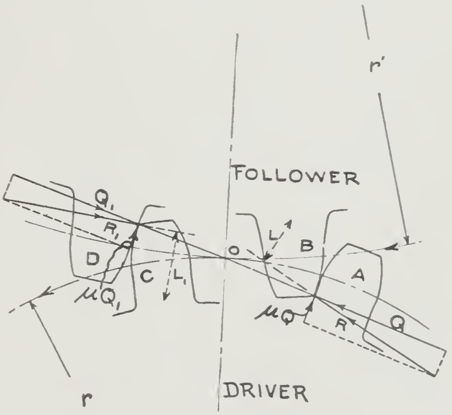


Fig. 33.

Tooth C of the DRIVER has a large bending moment, but small compressive and shearing stresses.

The conditions as noted above are not those of every pair of gears, in fact they vary with every difference of pitch circle, or of detail and position of tooth. It is true, however, that in nearly all cases in practice the bending stress is the controlling one from a theoretical standpoint. Moreover, the designer must consider the form and strength of the tooth when it is under the condition of maximum moment. This evidently, from the above, occurs at the beginning of contact, for the follower teeth; and at the end of contact, for the driver teeth. In the particular case illustrated in

Fig. 33, if the material in both gears were the same, tooth C, being the weaker at the root, would probably break before B; but if C were of steel, and B of cast iron, B might break first.

It will be noticed that R is nearly parallel to the top of the tooth; and it may easily happen that the friction may become of such a value that it will turn the direction of R until it lies along the top of the tooth exactly, which is the condition for maximum moment. For strength calculations it is usual to consider this condition as existing in all cases.

At the beginning of contact there is more or less shock when the teeth strike together, and this effect is much more evident at high speeds. There is also at the beginning of contact a sort of chattering action as the driving tooth rubs along the driven tooth.

Uniform distribution of pressure along the face of the tooth is often impaired by uneven wear of the bearings supporting the gear shafts, the pressure being localized on one corner of the tooth. The same effect is caused by the accidental presence of foreign material between the teeth. Again, in cast gearing, the spacing may be irregular, or, on account of draft on the pattern, the teeth may bear at the high points only. While it is usual to consider that the load is evenly distributed along the face of the tooth, yet the above considerations show that *an ample margin of strength must always be allowed* on account of these uncertainties.

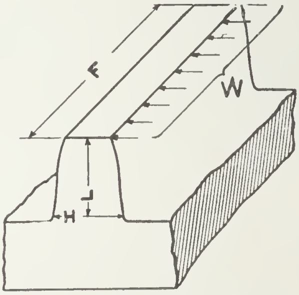


Fig. 34.

When the number of teeth in the mating gears is high, the load will be distributed between several teeth ; but, as it is almost certain that at some time the proper distribution of load will not exist, and that one tooth will receive the full load, it is considered that practically the only safe method is so to design the teeth that a single tooth may be relied upon to withstand the full load without failure.

THEORY. Based on the Analysis as given, the theory of gear teeth assumes that one tooth takes the whole load, and that this load is evenly distributed along the top of the tooth and acts parallel with its base, thus reducing the condition of the tooth to that of a cantilever beam. The magnitude of this load at the top of the tooth is taken for convenience the same as the force transmitted at the pitch circle. This condition is shown in Fig. 34. Equating the external moment to the internal moment, we then have, from mechanics:

$$WL = \frac{SI}{c} = \frac{SFH^2}{6}. \qquad (61)$$

The thickness H is usually taken either at the pitch line or at the root of the tooth just before the fillet begins; and L, of course, is dependent on the tooth dimensions. The formula is most readily used when the outline of the tooth is either assumed or known, a trial calculation being made to see if it will stand the load, and a series of subsequent calculations followed out in the same way until a suitable tooth is found. This method is pursued because there are certain even pitches which it is desirable to use; and it is safe to say that any calculation figured the reverse way would result in fractional pitches. The latter course may be used, however, and the nearest even pitch chosen as the proper one.

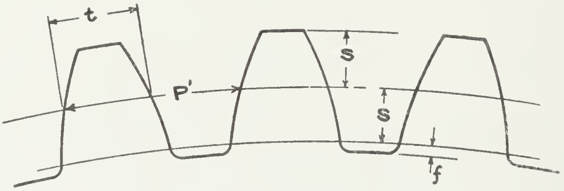
As stated under "Analysis," there are a great many circumstances attending the operation of gears which make impossible the purely theoretical application of the beam formulæ. For this reason there is no one element of machinery which depends so much on experience and judgment for correct proportion as the tooth of a gear. Hence it is true that a rational formula based on the theoretical one is really of the greater practical value in tooth design.

If we examine formula 61, we find that in a form solved for W, we have:

$$W = \frac{SFH^2}{6\,L}. \qquad (62)$$

Of these quantities, H and L are the only variables, for we can make the others what we choose. H and L depend upon the circular pitch P^1 and the curvature and outline of the tooth. If now we could settle on a standard system of teeth, we could establish a coefficient to be used to take the place of the variable part of H and L, which depends on the outline of tooth, and we shouid thus have an empirical formula which would be on a theoretical basis.

This, Mr. Wilfred Lewis has done; and it is safe to say that this formula is more universally used and with more satisfactory practical results than any other formula, theoretical or practical, that has ever been devised. His coefficient is known as y, and was determined from many actual drawings of different forms of teeth showing the weakest section. This coefficient is worked out for the three most common systems as follows:



$$t = \frac{P^1}{2} \qquad f = \frac{t}{10} \qquad s = \frac{P^1}{\pi}$$

Fig. 35.

For 20° involute, $$y = 0.154 - \frac{0.912}{N}. \qquad (63)$$

For 15° involute and cycloidal, $$y = 0.124 - \frac{0.684}{N}. \qquad (64)$$

For radial flanks, $$y = 0.075 - \frac{0.276}{N}. \qquad (65)$$

The tooth upon which the above is based is the American standard or Brown & Sharpe tooth, for which the proportions are shown in Fig. 35.

The "Lewis" formula* is:

$$W = SP^1 Fy. \qquad (66)$$

A table indicating the value of S for different speeds follows:

Safe Working Stresses for Different Speeds.

Speed of teeth, ft. per min.	100	200	300	600	900	1200	1800	2400
Cast iron	8000	6000	4800	4000	3000	2400	2000	1700
Steel	20000	15000	12000	10000	7500	6000	5000	4300

---

*NOTE. A full and convenient statement of the Lewis formula will be found in "Kent's Pocket Book."

A usual relation of F to P^1 is:

$$\text{For cast teeth, } F = 2P^1 \text{ to } 3P^1. \qquad (66)$$

$$\text{For cut teeth, } F = 3P^1 \text{ to } 4P^1. \qquad (67)$$

The usual method of handling these formulæ is as follows:

The pitch circles of the proposed gears are known or can be assumed; hence W can readily be figured, also the speed of the teeth, whence S can be read from the table. The desired relation of F to P^1 can be arbitrarily chosen, when P^1 and y become the only unknown quantities in the equation. A shrewd guess can be made for the number of teeth, and y calculated therefrom. Then solve the equation for P^1 which will undoubtedly be fractional. Choose the nearest even pitch, or, if it is desired to keep an even diametral pitch, the fractional pitch that will bring an even diametral pitch. Now, from this final and corrected pitch, and the diameter of the pitch circle, calculate the number of teeth N in the gear. Check the assumed value of y by this positive value of N.

Another good way of using this formula is to start with the pitch and face desired, and the diameter of the pitch circle. In this case W is the only unknown quantity, and when found can be compared with the load required to be carried. If too small, make another and successive calculations until the result approximates the required load.

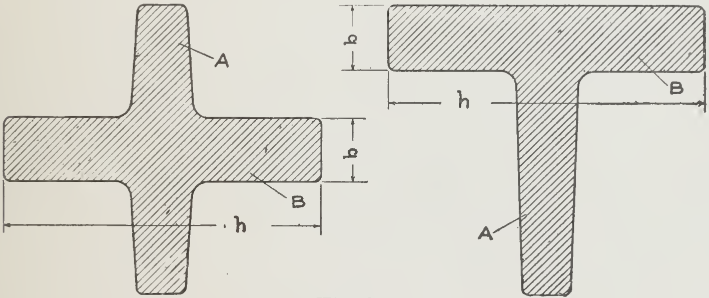


Fig. 37.

SPUR GEAR RIM, ARMS, AND HUB.

ANALYSIS and THEORY. The rim of a gear has to transmit the load on the teeth to the arms. It is thus in tension on one side of the teeth in action, and in compression on the other. The section of the rim, however, is so dependent on other practical considerations which call for an excess of strength in this respect, that it is not considered worth while to attempt a calculation on this basis.

Gears seldom run fast enough to make necessary a calculation for centrifugal force ; and in general it can be said that the design of the rim is entirely dependent on practical considerations. These will appear later under " Practical Modification."

The arms of a gear are stressed the same as pulley arms, the same theory answering for both, except that a gear rim always being much heavier than a pulley rim, the distribution of load amongst the arms is better in the case of a gear than of a pulley, and it is usually safe to assume that each arm of a gear takes its full proportion of load ; or, for an oval section, equating the external moment to the internal moment as in the case of pulleys, we have :

$$\frac{WD}{n2} = 0.0393\ Sh^3. \qquad (68)$$

Heavy spur gears have the arms of a cross or T section (Fig. 37), the latter being especially applicable to the case of bevel gears where there is considerable side thrust. The simplest way of treating such sections is to consider that the whole bending moment is taken by the rectangular section whose greater dimension is in the direction of the load. The rest of the section, being close to the neutral axis of the section, is of little value in resisting the direct load, its function being to give sidewise stiffness. The equation for the cross or T style of arm, then is :

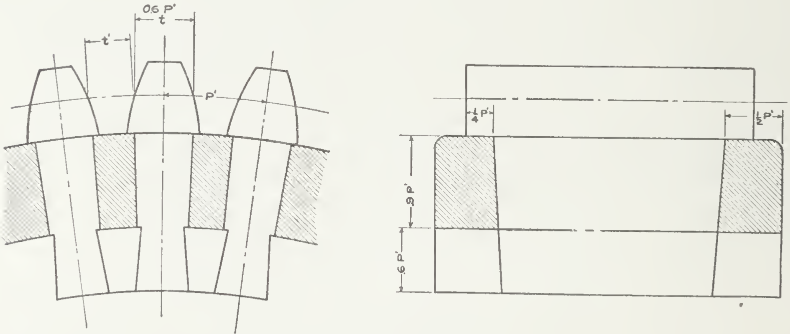


Fig. 38.

$$\frac{W}{n} \times \frac{D}{2} = \frac{Sbh^2}{6}. \qquad (69)$$

Either b or h may be assumed, and the other determined. As a guide to the section, b may be taken at about the thickness of the tooth.

Gear hubs are in no wise different from the hubs of pulleys or other rotating pieces. The depth necessary for providing sufficient strength over the key to avoid splitting is the guiding element, and can usually be best determined by careful judgment.

PRACTICAL MODIFICATION. The practical requirements, which no theory will satisfy, are many and varied. Sudden and severe shock, excessive wear due to an atmosphere of grit and corrosive elements, abrupt reversal of the mechanism, the throwing-in of clutches and pawls, the action of brakes—these and many other influences have an important bearing on gear design, but not one that can be calculated. The only method of procedure in such cases is to base the design on analysis and theory as previously given, and then add to the face of gear, thickness of tooth, or pitch an amount which judgment and experience dictate as sufficient.

Excessive noise and vibration are difficult to prevent at high speeds. At 1,000 feet per minute, gears are apt to run with an unpleasant amount of noise. At speeds beyond this, it is often necessary to provide **mortise teeth,** or teeth of hard wood set into a cast-iron rim (see Fig. 38). Rawhide pinions are useful in this regard. Fine pitches with a long face of tooth run much more smoothly at high speeds than a coarse pitch and narrow-faced tooth of equal strength. Greater care in alignment of shafts, however, is necessary, also stiffer supports.

Should it be impracticable to use a standard tooth of sufficient strength, there are several ways in which we can increase the carrying capacity without increasing the pitch. These are:

1. Use a stronger material, such as steel.
2. Shroud the teeth.
3. Use a hook tooth.
4. Use a stub tooth.

Shrouding a tooth consists in connecting the ends of the teeth with a rim of metal. When this rim is extended to the top of the tooth, the process is called "full-shrouding" (Fig. 39); and when carried only to the pitch line, it is termed "half-shrouding" (Fig. 40). The theoretical effect of shrouding is to make the tooth act like a short beam built in at the sides; and the tooth will practically have to be sheared out in order to fail. This modification of gear design requires the teeth to be cast, as the cutter cannot pass through the shrouding. The strength of the shrouded gear is estimated to be from 25 to 50 per cent above that of the plain-tooth type.

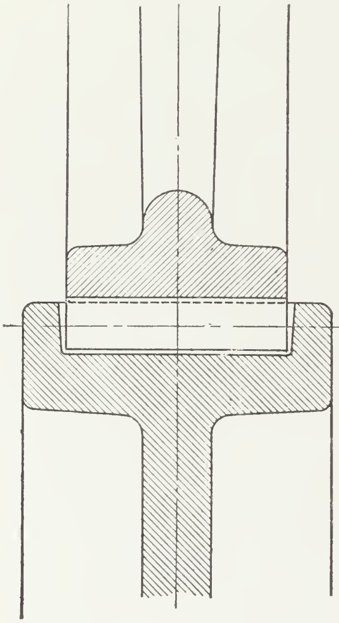


Fig. 39.

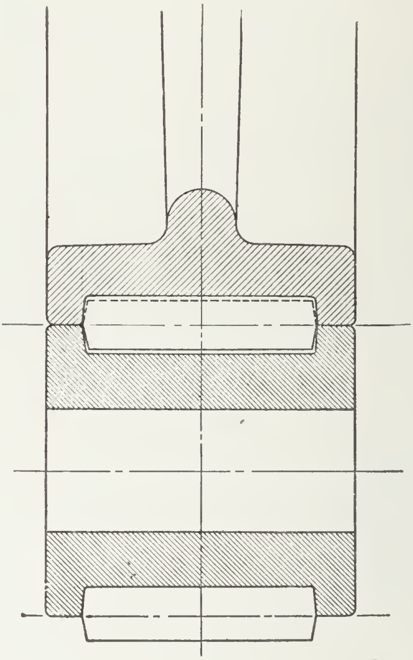


Fig. 40.

The **hook-tooth gear** (Fig. 41) is applicable only to cases where the load on the tooth does not reverse. The working side of the tooth is made of the usual standard curve, while the back is made of a curve of greater obliquity, resulting in a considerable increase of thickness at the root of the tooth. A comparison of strength between this form and the standard may be made by drawing the two teeth for a given pitch, measuring their thickness just at top of the fillet, and finding the relation of the squares of these dimensions. The truth of this relation is readily seen from an inspection of formula 61.

The **stub tooth** merely involves the shortening of the height of the tooth in order to reduce the lever arm on which the load acts, thus reducing the moment, and thereby permitting a greater load to be carried for the same stress.

The rim of a gear is dependent for its proportions chiefly on questions of practical moulding and machining. It must bear a certain relation to the teeth and arms, so that, when it is cooling in the mould, serious shrinkage stresses will not be set up, forming pockets and cracks. Moreover, when under pressure of the cutter in the producing of the teeth, it must not chatter or spring. This condition is quite well attained in ordinary gears when the thickness of the rim below the base of the tooth is made about the same as the thickness of the tooth.

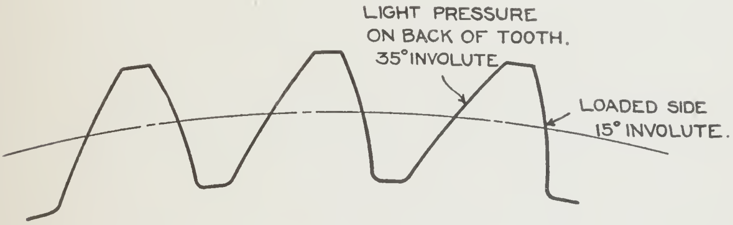


Fig. 41.

The stiffening ribs and arms must all be joined to the rim by ample fillets, and the cross-section must be as uniform as possible, to prevent unequal cooling and consequent pulling-away of the arms from the rim or hub. Often the calculated size of the arms at both rim and hub has to be modified considerably to meet this requirement.

The arms are usually tapered to suit the designer's eye, a small gear requiring more taper per foot than a large one. Both rim and hub should be tapered $\frac{1}{2}$ inch per foot to permit easy drawing-out from the mould.

The proportions given in the following table have been used with success as a basis of gear design in manufacturing practice. The table will serve as an excellent guide in laying out, and can be closely followed, in most cases with but slight modification. **Web gears** are introduced for small diameters where the arms begin to look awkward and clumsy.

Gear Design Data.

Measurements given in inches. Letters refer to Fig. 42.

Diametral pitch ..	P	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5	6	8
Face............	F	$6\frac{1}{4}$	$5\frac{1}{2}$	$4\frac{3}{4}$	$3\frac{3}{4}$	$3\frac{1}{4}$	$2\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{7}{8}$	$1\frac{1}{2}$
Thickness of arm when extended to pitch line....	G	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{1}{2}$
Width of arm when extended to pitch line	C	4	$3\frac{1}{2}$	3	$2\frac{1}{2}$	$2\frac{1}{4}$	2	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{8}$
Thickness of rim..	K	$2\frac{3}{4}$	$2\frac{3}{8}$	$2\frac{1}{8}$	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	1	$\frac{7}{8}$	$\frac{3}{4}$
Depth of rib......	E	2	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	1	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{8}$
Thickness of web.	T	$1\frac{1}{8}$	1	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{5}{16}$

Number of arms, 6.

Give inside of rims and hub a draft of $\frac{1}{2}$ inch per foot.

BEVEL GEARS.

NOTATION—The following notation is used throughout the chapter on Bevel Gears:

A =Apex distance at pitch element of cone (inches).
A^1=Apex distance at bottom element of tooth (inches).
B =Angle of bottom of tooth (degrees).
C =Pitch angle (degrees).
D =Pitch diameter (inches).
E =Radius increment of gear (inches).
F =Face of gear (inches).
f =Clearance at bottom (inches).
G =Angle of face (degrees).
H =Cutting angle (degrees).
K =Radius increment of pinion (inches).
N =Number of teeth.
N^1=Formative number of teeth, or the number corresponding to the spur gear on which the outline of tooth is made.
O D=Outside diameter (inches).
P =Diametral pitch related to pitch diameter (teeth per inch).
P^1 =Circular pitch measured on the circumference of D (inches).
S =Working strength of material (lbs. per sq. in.).
s =Addendum, or height of tooth above pitch line (inches).
$s+f$=Depth of tooth below pitch line (inches).
T =Angle of top of tooth (degrees).
t =Thickness of tooth at pitch line (inches).
W =Working load at pitch line (lbs.).
y =Factor in "Lewis" formula.

ANALYSIS. It is possible to consider bevel gears as the general case of which spur gears are a special form. The pitch surfaces of spur gears described above as cylinders, mathematically considered, are cones whose vertices are infinitely distant, while bevel gears likewise are based on pitch cones, but with a vertex at some finite point, common to the mating pair. Hence, as we might expect, the laws of tooth action are similar in bevel gears to those in the case of spur gears. The profile of the tooth in the former case, however, is based, not on the real radius of the pitch cone, but on the radius of the normal cone; and in the development of the outline the latter is treated just as though it were the radius of a spur gear. The tooth thus formed is wrapped back upon the normal cone face, and becomes the large end of the tapering bevel-gear tooth (see Fig. 44).

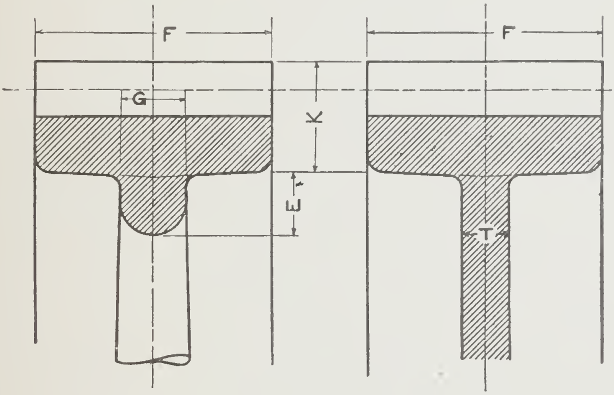


Fig. 42.

The teeth of bevel gears, being simply projections with bases on the pitch cones, have a varying cross-section decreasing toward the vertex; also a trapezoidal section of root, the latter section acting as a beam section to resist the cantilever moment due to the tooth load.

The arms must, as in the case of spur gears, transmit the load from the tooth to the shaft; in addition, the arms of a bevel gear are subjected to a side thrust due to the wedging action of the cones. Hence sidewise stiffness of the arms is more essential in this type of gear than in the case of the spur gear.

THEORY. It is evident that the calculation of tooth strength based on a trapezoidal section of root would be somewhat compli-

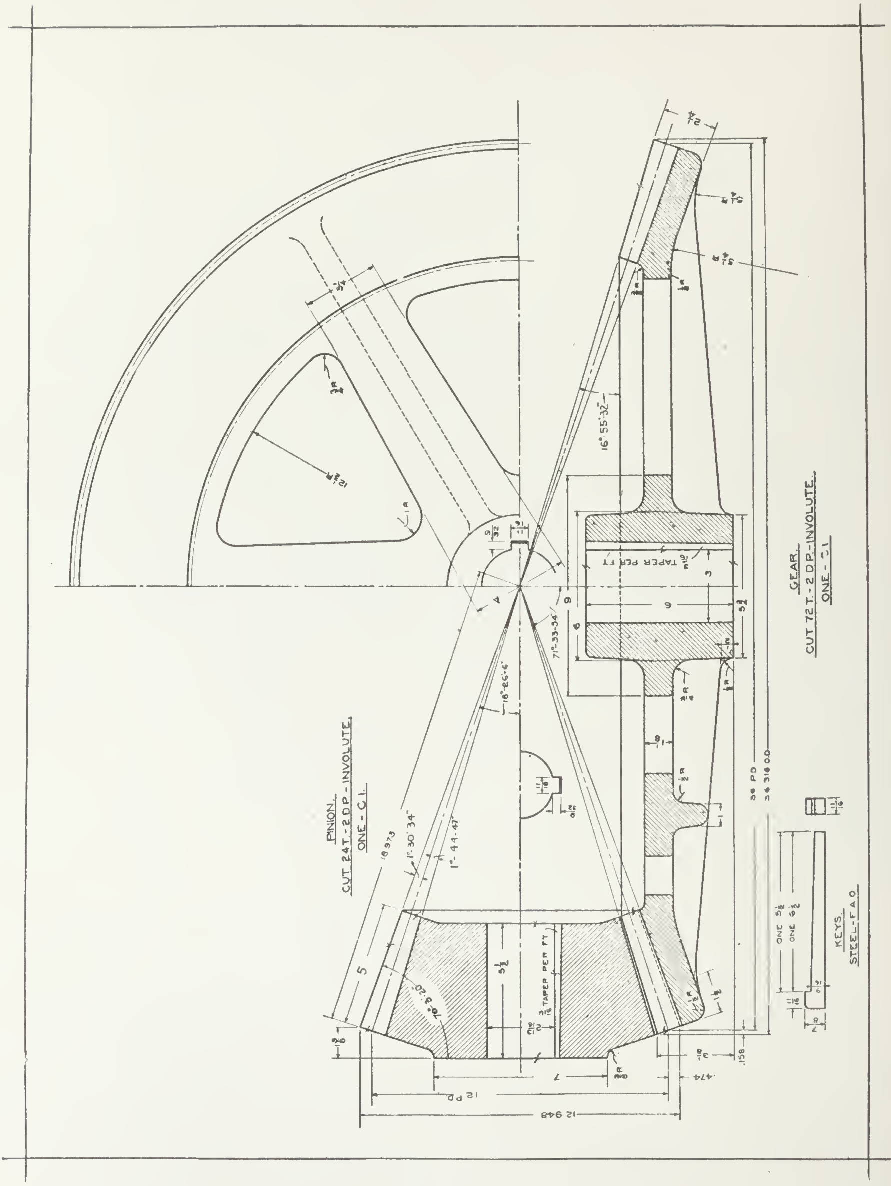


Fig. 43.

cated; also that the trapezoid in most cases would be but little different from a true rectangle. Hence the error will be but slight if the average cross-section of the tooth be taken to represent its strength, and the calculation made accordingly.

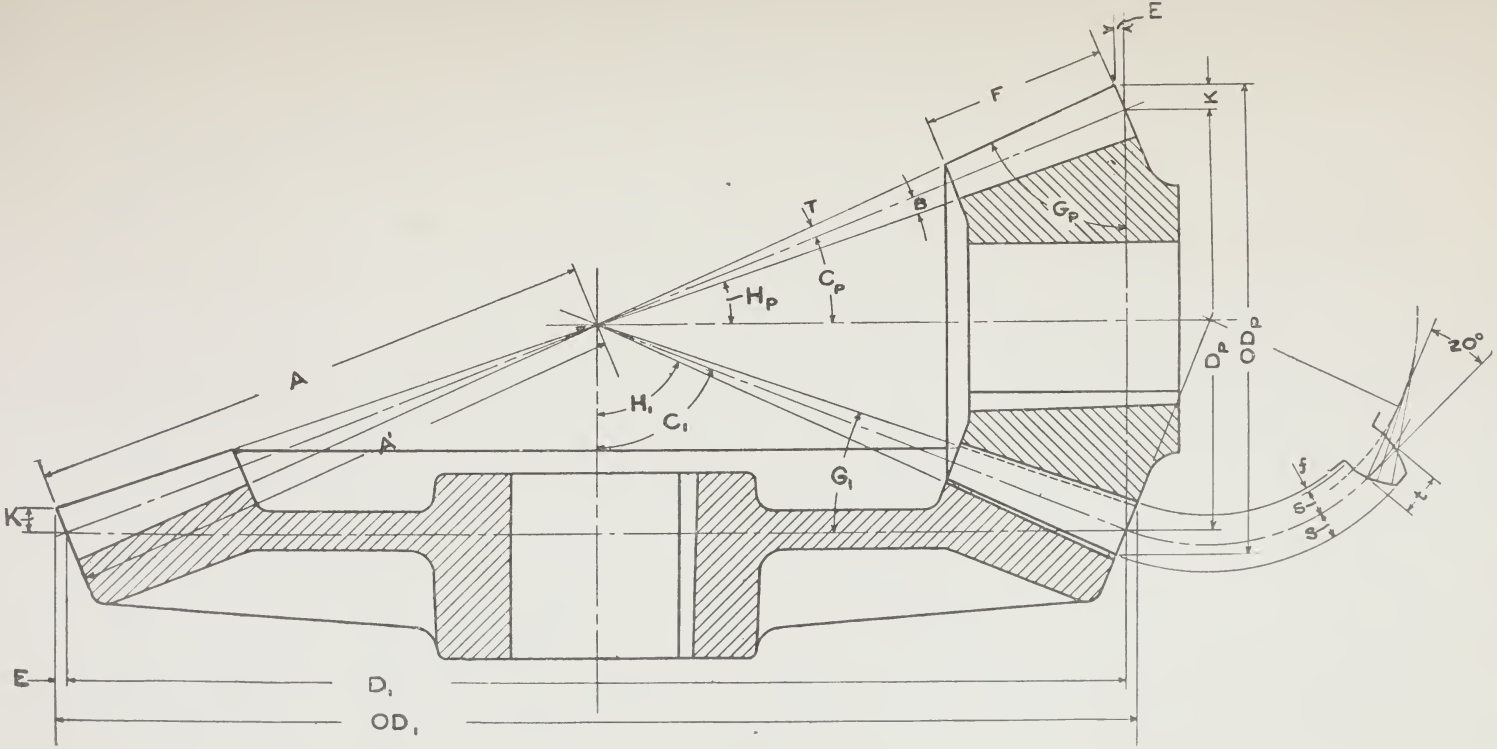


Fig. 44.

Fig. 45 shows a bevel-gear tooth with the average cross-section in dotted lines. For the purpose of calculation, the assumption is made that the section A is carried the full length of the face of the gear, and that the load which this average tooth must carry is the calculated load at the pitch line of section A. This is equivalent to saying that the strength of a bevel-gear tooth is equal to that of a spur-gear tooth which has the same face, and a section identical with that cut out by a plane at the middle of the bevel tooth. The load, as in the case of the spur gear, should be taken at the top of the tooth; and its magnitude can be conveniently calculated at the mean pitch radius of the bevel face, without appreciable error.

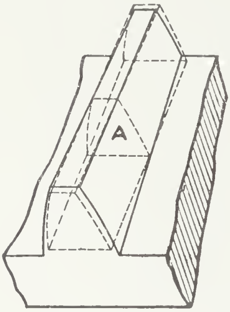


Fig. 45.

This similarity to spur gears being borne in mind, the calculation for strength needs no further treatment. Once the average tooth is assumed or found by layout, a strict following-out of the methods pursued for spur-gear teeth will bring consistent results.

The detail design of a pair of bevel gears involves some trigonometrical computations in order properly to dimension the drawing for use in finishing the blanks and subsequently in cutting the teeth, or, in the case of cast gears, in making the pattern. These calculations, although simple, are yet apt to be tedious; and inaccuracies are likely to creep in if a definite system of relations be not maintained. Hence the results of these calculations are given below in condensed and reduced form. The deduction of these formulæ is a simple and interesting exercise in trigonometry; and it is urged that they be worked out by the student from the figure, in which case he will feel greater confidence in their use.

Axes of Gears at 90 Degrees.

Use subscript 1 for gear; P for pinion. Letters refer to Fig. 44.

$$P = \frac{N}{D} = \frac{\pi}{P^1}. \qquad (70)$$

$$s = \frac{1}{P} = \frac{P^1}{\pi}. \qquad (71)$$

$$t = \frac{P^1}{2} = \frac{\pi}{2P}. \qquad (72)$$

$$f = \frac{t}{10} = \frac{P^1}{20} = \frac{\pi}{20P}. \tag{73}$$

$$\tan C_p = \frac{N_p}{N_1};\ \tan C_1 = \frac{N_1}{N_p}. \tag{74}$$

$$\tan T = \frac{s}{A} = \frac{2 \sin C}{N}. \tag{75}$$

$$\tan B = \frac{s+f}{A} = \frac{2.314 \sin C}{N}. \tag{76}$$

$$s + f = A \tan B = \frac{1.157}{P} = 0.368P^1. \tag{77}$$

$$A = \frac{N}{2P \sin C} = \frac{1}{2P}\sqrt{N_1^2 + N_p^2} = \frac{1}{2}\sqrt{D_1^2 + D_p^2}. \tag{78}$$

$$A^1 = \frac{A}{\cos B} = \frac{N}{2P \cos B \sin C}. \tag{79}$$

$$G_1 = 90° - (C_1 + T);\ G_p = 90° - (C_p + T). \tag{80}$$

$$E = S \cos C_1 = S \sin C_p. \tag{81}$$

$$K = S \cos C_p = S \sin C_1. \tag{82}$$

PRACTICAL MODIFICATION. The practical requirements to be met in transmission of power by bevel gears are the same as for spur gears; but in the case of bevel gears even greater care is necessary to provide stiffness, strength, true alignment, and rigid supports. As far as the gears themselves are concerned, a long face is desirable; but it is much more difficult to gain the advantage of its strength than in the case of spur gears, because full bearing along the length of the tooth is hard to guarantee.

The rim usually requires a series of ribs running to the hub to give required stiffness and strength against the side thrust which is always present in a pair of bevel gears. Instead of arms, the tendency of bevel-gear design, except for very large gears, is toward a web on account of the better and more uniform connection thereby secured between rim and hub. This web may be lightened by a number of holes, so that the resultant effect is that of a number of wide and flat arms.

The hubs naturally have to be fully as long as those of spur gears, because there is greater tendency to rock on the shaft, due to the side thrust from the teeth, mentioned above.

The teeth on small gears are cut with rotary cutters, at least two finishing cuts being necessary, one for each side of the tapering tooth. The more accurate method is to plane the teeth on a special gear planer, and this method is followed on all gears of any considerable size. The practical requirement here is that no portion of the hub shall project so as to interfere with the stroke of the planer tool. The requirements of gear planers vary somewhat in this regard.

Finally, after all that is possible has been done in the design of the gear itself to render it suitable to withstand the varied stresses, especial attention must be paid to the rigidity of the supporting shafts and bearings. Bearings should always be close up to the hubs of the gears, and, if possible the bearing for both pinion and gear should be cast in the same piece. If this is not done, the tendency of the separate bearings to get out of line and destroy the full bearing of the teeth is difficult to control. Thrust washers are desirable against the hubs of both pinion and gear; also proper means of well lubricating the same.

With these considerations carefully met, bevel gears are not the bugbear of machine design that they are sometimes claimed to be. The common reason why bevel gears cut and fail to work smoothly, is that the gears and supports are not designed carefully enough in relation to each other. This is also true of spur gears, but the bevel gear will reveal imperfections in its design far the more quickly of the two.

WORM AND WORM GEAR.

NOTATION—The following notation is used throughout the chapter on Worm and Worm Gear:

D = Pitch diameter of gear (inches).
E = Efficiency between worm shaft and gear shaft (per cent).
f = Clearance of tooth at bottom (inches).
i = Index of worm thread (1 for single, 2 for double, etc.).
L = Lead of worm thread (inches).
M = Revolutions of gear shaft per minute.
M_w = Revolutions of worm shaft per minute.
N = Number of teeth in gear.
P^1 = Circular pitch = Pitch of worm thread (inches).
R = Radius of pitch circle of worm gear (inches).
s = Addendum of tooth (inches).
T = Twisting moment on gear shaft (inch-lbs.).
T_w = Twisting moment on worm shaft (inch-lbs.).
t = Thickness of tooth at pitch line (inches).
W = Load at pitch line (lbs.).

ANALYSIS. The simplest way of analyzing the case of the worm and worm gear is to base it upon an ordinary screw and nut. Take, for example, the lead screw of a common lathe. The carriage carries a nut, through which the lead screw passes. By the rotation of the screw, the carriage, being constrained by the guides to travel lengthwise of the ways, is moved. This motion is, for a single-threaded screw, a distance per revolution equal to the lead of the screw.

Now, suppose that the carriage, instead of sliding along the ways, is compelled to turn about an axis at some point below the ways. Also, suppose the top of the nut to be cut off, and its length made endless by wrapping it around a circle struck from the center about which the carriage rotates. This reduces the nut to a peculiar kind of spur gear, the partial threads of the nut now having the appearance of twisted teeth.

This special form of spur gear, based on the idea of a threaded nut, is known as a **worm gear,** and the screw is termed a **worm.** The teeth are loaded similarly to those of a spur gear, but with the additional feature of a large amount of sliding along the tooth surfaces. This, of course, means considerable friction; and it is in fact possible to utilize the worm and worm gear as an efficient device, only by running the teeth constantly in a bath of oil. Even then the pressures have to be kept well down to insure the required term of life of the tooth surfaces.

It is evident that for one revolution of a single-threaded worm, one tooth of the gear will be passed. The speed ratio between the worm gear and worm shaft will then be equal to the number of teeth in the gear, which is relatively great. Hence the worm and worm gear are principally useful in giving large speed reduction in a small amount of space.

THEORY. The theory of worm-wheel teeth is complicated and obscure. The production of the teeth is simple, a dummy worm with cutting edges, called a "hob," being allowed to carve its way into the worm-gear blank, thus producing the teeth and at the same time driving the worm gear about its axis.

It is clear that if we know the torsional moment on the worm-gear shaft, and the pitch radius of the worm gear, we can find the load on the teeth at the pitch line by dividing the former by the latter. Expressed as an equation:

$$WR = T; \text{ or } W = \frac{T}{R}. \qquad (83)$$

How we shall consider this value of W as distributed on the teeth, is a question difficult to answer. The teeth not only are curved to embrace the worm, but are twisted across the face of the gear, so that it would be practically impossible to devise a purely theoretical method of exact calculation. The most reasonable thing to do is to assume the teeth as being equally as strong as spur-gear teeth of the same circular pitch, and to figure them accordingly. It is probably true, however, that the load is carried by more than one tooth, especially in a hobbed wheel; so we shall be safe in assuming that two—and, in case of large wheels, three—teeth divide the load between them. With these considerations borne in mind, the case reduces itself to that of a simple spur-gear tooth calculation, which has already been explained under the heading "Spur Gears."

The worm teeth, or threads, are probably always stronger than the worm-gear teeth; so no calculation for their strength need be made.

The twisting moment on the worm shaft is not determined so directly as in the case of spur gears. The relative number of revolutions of the two shafts depends upon the "lead" of the worm thread and the number of teeth in the gear.

Lead (L) is the distance parallel to the axis of the worm which any point in the thread advances in one revolution of the worm. **Pitch** (P^1) is the distance parallel to the axis of the worm between corresponding points on adjacent threads. The distinction between lead and pitch should be carefully observed, as the two are often confounded, one with the other.

The thread may be single, double, triple, etc., the index of the thread i, being 1, 2, 3, etc., in accordance therewith. The relation between lead and pitch may then be expressed by an equation, thus:

$$L = i\, P^1. \qquad (84)$$

When the index of the thread is changed the speed ratio is changed, the relation being shown by the equation:

$$\frac{M}{M_w} = \frac{i}{N} \qquad (85)$$

If the efficiency were 100 per cent between the two shafts, the twisting moments would be inversely as the ratio of the speeds thus:

$$\frac{T_w}{T} = \frac{M}{M_w} = \frac{i}{N};$$

or,
$$T_w = \frac{Ti}{N}; \qquad (86)$$

but for an efficiency E the equation would be:

$$\frac{T_w}{T} = \frac{i}{EN};$$

or,
$$T_w = \frac{Ti}{EN}. \qquad (87)$$

The diameter of the worm is arbitrary. Change of this diameter has no effect on the speed ratio. It has a slight effect on the efficiency, the smaller worm giving a little higher efficiency. The diameter of the worm runs ordinarily from 3 to 10 times the circular pitch, an average value being $4P^1$ or $5P^1$.

A longitudinal cross-section through the axis of the worm cuts out a rack tooth, and this tooth section is usually made of the standard $14\frac{1}{2}°$ involute form shown in Fig. 46 for a rack.

The end thrust, of a magnitude practically equal to the pressure between the teeth, has to be taken by the hub of the worm against the face of the shaft bearing. A serious loss of efficiency from friction is likely to occur here. This is often reduced, however, by roller or ball bearings. With two worms on the same shaft, each driving into a separate worm gear, it is possible to make one of the worms right-hand thread, and the other left-hand, in which case the thrust is self-contained in the shaft itself, and there is absolutely no end thrust against the face of the bearing. This involves a double outfit throughout, and is not always practicable.

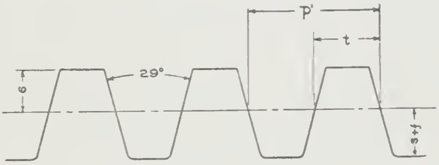


Fig. 46.

There are few mathematical equations necessary for the dimensioning of a worm and worm gear. The formulæ for the tooth parts as given on page 120 apply equally well in this case.

PRACTICAL MODIFICATION. The discussion of the efficiency E of the worm and worm gear is more of a practical than of a theoretical nature. It seems to be true from actual operation, as well as theory, that the steeper the threads the higher the efficiency. In actual practice we seldom have opportunity to change the slope of the thread to get increased efficiency. The slope is usually settled from considerations of speed ratio, or available space, or some other condition. The usual practical problem is to take a given worm and worm gear, and to make out of it as efficient a device as possible. With hobbed gears running in oil baths, and with moderate pressures and speeds, the efficiency will range between 40 per cent and 70 per cent. The latter figure is higher than is usually attained.

To avoid cutting and to secure high efficiency, it seems essential to make the worm and the gear of different materials. The worm-thread surfaces being in contact a greater number of times than the gear teeth, should evidently be of the harder material. Hence we usually find the worm of steel, and the gear of cast iron, brass, or bronze. To save the expense of a large and heavy bronze gear, it is common to make a cast-iron center and bolt a bronze rim to it.

The worm, being the most liable to replacement from wear, it is desirable so to arrange its shaft fastening and general accessibility that it may be readily removed without disturbing the worm gear.

The circular pitch of the gear and the pitch of the worm thread must be the same, and the practical question comes in as to the threads per inch possible to be cut in the lathe in the production of the worm thread. The pitch must satisfy this requirement; hence the pitch will usually be fractional, and the diameter of the worm gear, to give the necessary number of teeth, must be brought to it. While it would perhaps be desirable to keep an even diametral pitch for the worm gear, yet it would be poor design to specify a worm thread which could not be cut in a lathe.

The standard involute of $14\frac{1}{2}°$, and the standard proportions of teeth as given on page 120, are usually used for worm threads. This system requires the gear to have at least 30 teeth, for if fewer teeth are used the thread of the worm will interfere with the flanks of the gear teeth. This is a mathematical relation, and there are methods of preventing it by change of tooth proportions or of angle of worm thread ; but there are few instances in which less than 30 teeth are required, and it is not deemed worth while to go into a lengthy discussion of this point.

The angle of the worm embraced by the worm-gear teeth varies from 60° to 90°, and the general dimensions of rim are made about the same as for spur gears. The arms, or the web, have the same reasons for their size and shape. Probably web gears and cross-shaped arms are more common than oval or elliptical sections.

Worm gears sometimes have cast teeth, but they are for the roughest service only, and give but a point bearing at the middle of the tooth. An accurately hobbed worm gear will give a bearing clear across the face of the tooth, and, if properly set up and cared for, makes a good mechanical device although admittedly of somewhat low efficiency.

Fig. 47 shows a detail drawing of a standard worm and worm gear. It should serve as a suggestion in design, and an illustration of the shop dimensions required for its production.

PROBLEMS ON SPUR, BEVEL, AND WORM GEARS.

1. Calculate proportions of a standard Brown & Sharpe gear tooth of $1\frac{1}{2}$ diametral pitch, making a rough sketch and putting the dimensions on it.

2. Suppose the above tooth to be loaded at the top with 5,000 lbs. If the face be 6 inches, calculate the fiber stress at the pitch line, due to bending.

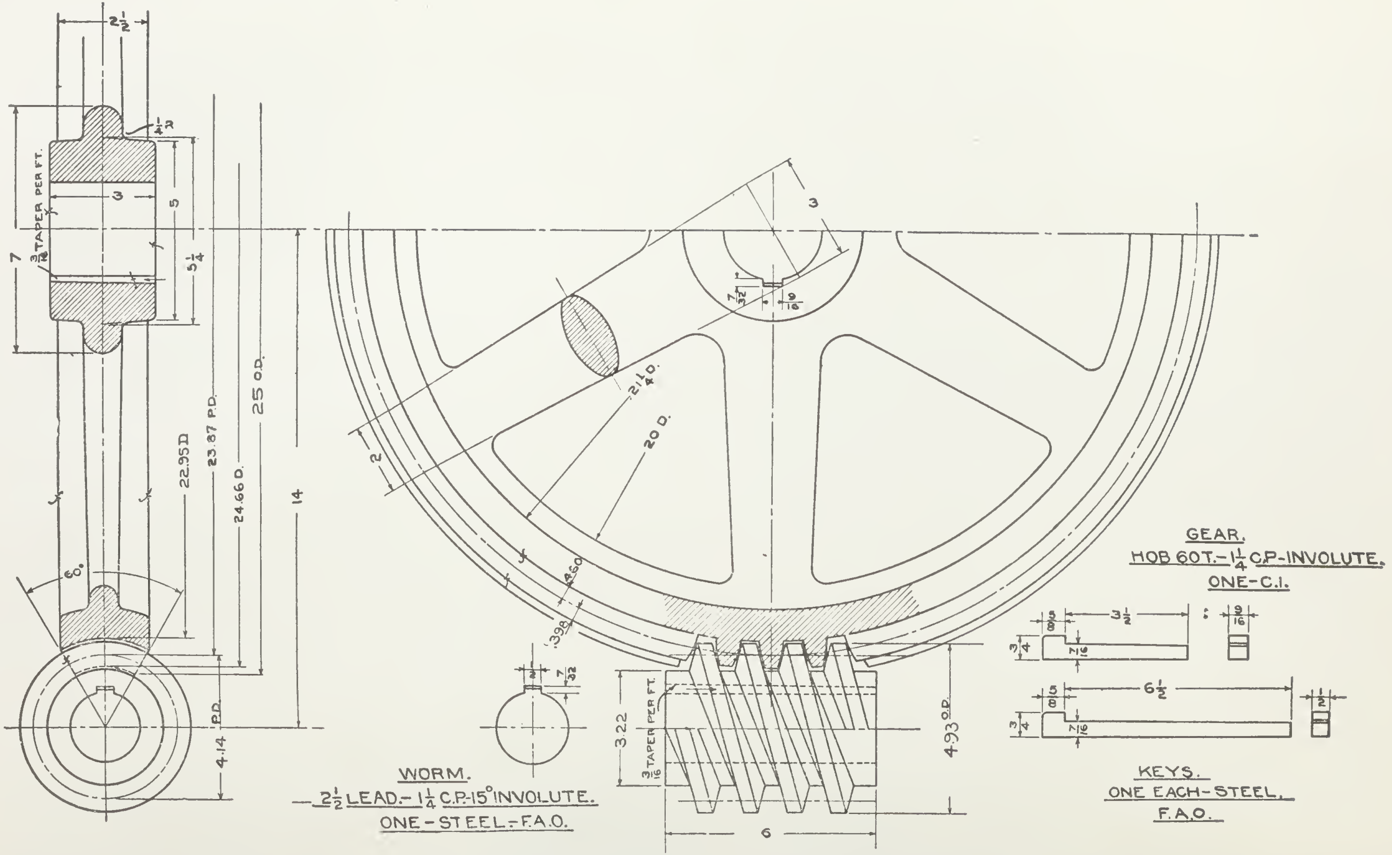
3. A tooth load of 1,200 lbs. is transmitted between two spur gears of 12-inch and 30-inch diameter, the latter gear making 100 revolutions per minute. Calculate a suitable pitch and face of tooth by the "Lewis" formula.

4. Assuming a $\frac{1}{2}$-inch web on the 12-inch gear, calculate the shearing fiber stress at the outside of a hub 4 inches in diameter

5. Design elliptical arms for the 30-inch gear, allowing $S = 2,200$.

6. Design cross-shaped arms for 30-inch gear.

7. Calculate the dimensions shown in formulæ 70 to 82 inclusive for a pair of bevel gears of 20 and 60 teeth respectively, 2 diametral pitch, and 4-inch face. (The use of logarithmic tables makes the calculation much easier than with the natural functions.)



SECTION AND ELEVATION OF WORM GEAR.
Fig. 47

8. A worm wheel has 40 teeth, 3 diametral pitch, and double thread. Calculate (*a*) its lead; (*b*) its pitch diameter.

FRICTION CLUTCHES.

NOTATION—The following notation is used throughout the chapter on Friction Clutches:.

α =Angle between clutch face and axis of shaft (degrees)
H =Horse-power (33,000 ft.-lbs. per minute).
μ =Coefficient of friction (per cent).
N =Number of revolutions per minute.
P =Force to hold clutch in gear to produce W (lbs.).
R =Mean radius of friction surface (inches).
T =Twisting moment about shaft axis (inch-lbs.).
V =Force normal to clutch face (lbs.).
W =Load at mean radius of friction surface (lbs.).

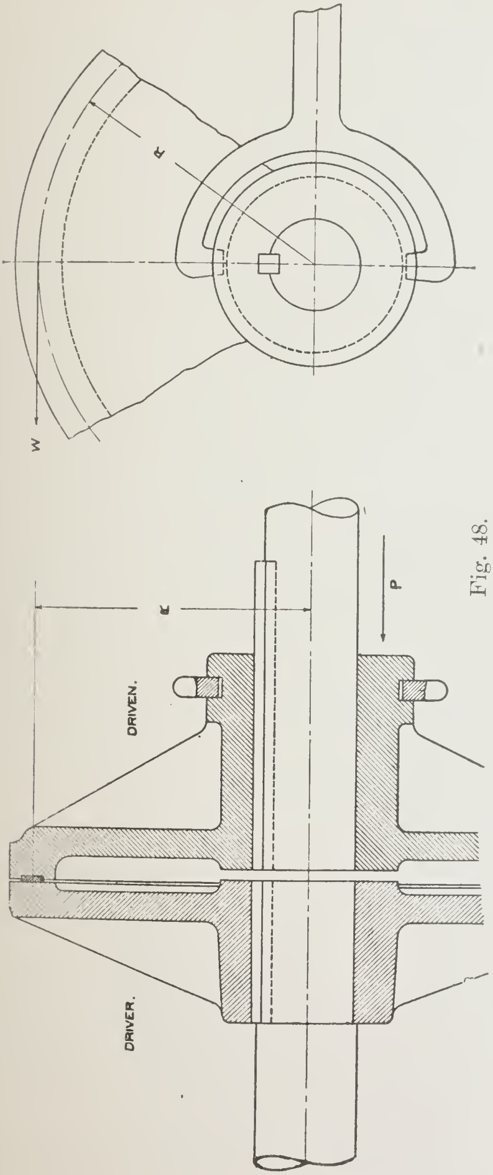


Fig. 48.

ANALYSIS. The friction clutch is a device for connecting at will two separate pieces of shaft, transmitting an amount of power between them to the capacity of the clutch. The connection is usually accomplished while the driving shaft is under full speed, the slipping between the surfaces which occurs during the throwing-in of the clutch, permitting the driven shaft to pick up the speed of the other gradually, without appreciable shock. The disconnection is made in the same manner, the amount of slipping which occurs depending on the suddenness with which the clutch is thrown out.

The force of friction is the sole driving element, hence the problem is to secure as large a force of friction as possible. But friction cannot be secured without a heavy normal pressure between surfaces having a high coefficient of friction between them. The many varieties of friction clutches which are on the market or designed for some special purpose, are all devices for accomplishing one and the same effect, *viz.*, the production of a heavy normal force or pressure between surfaces at such a radius from the driven axis, that the product of the force of friction thereby created and the radius shall equal the desired twisting moment about that axis.

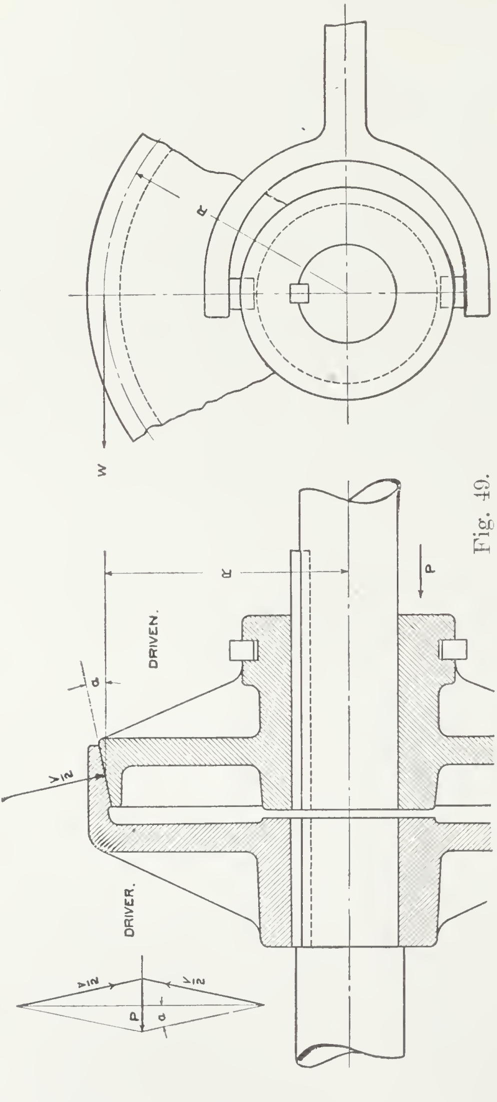


Fig. 49.

Three typical methods of accomplishing this are shown in Figs. 48, 49, and 50. None of these drawings is worked out in operative detail. They are merely illustrations of principle, and are drawn in the simplest form for that purpose.

In Fig. 48 the normal pressure is created in the simplest possible way, an absolutely direct push being exerted between the discs, due to the thrust P of the clutch fork.

In Fig. 49 advantage is taken of the wedge action of the inclined faces, the result being that it takes less thrust P to produce the required normal pressure at the radius R.

In Fig. 50 the inclination of the faces is carried so far that the angle *a* of Fig. 49 has become zero; and by the toggle-joint action of the link pivoted to the clutch collar, the normal force produced may be very great for a slight thrust P. By careful adjustment of the length of the link so that the jaw takes hold of the clutch surface, when the link stands nearly vertical, a very easy operating device is secured, and the thrust P is made a minimum.

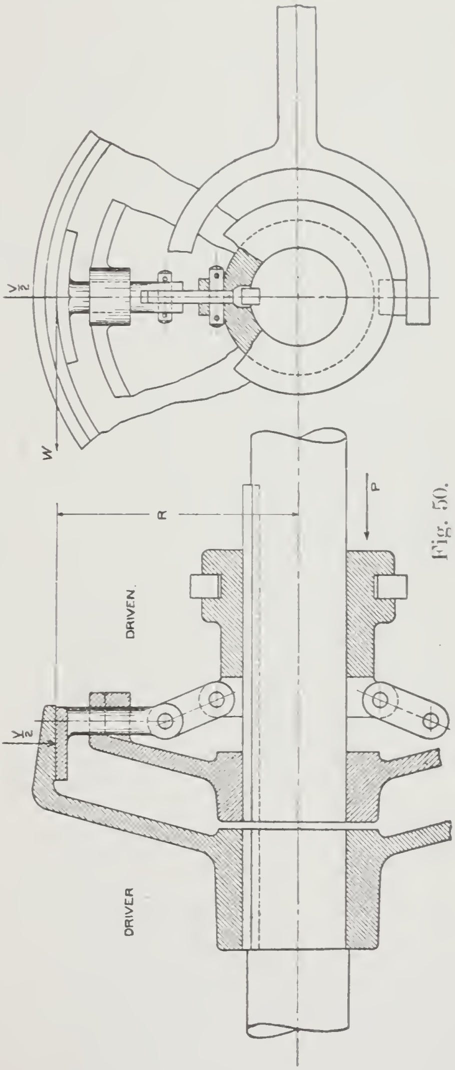


Fig. 50.

THEORY. Referring to Fig. 48 in order to calculate the twisting moment, we must remember that the force of friction between two surfaces is equal to the normal pressure times the coefficient of friction. This, in the form of an equation, using the symbols of the figure, is:

$$W = \mu P. \qquad (88)$$

Hence we may consider that we have a force of magnitude μP acting at the mean radius R of the clutch surface. The twisting moment will then be :

$$T = WR = \mu PR. \qquad (89)$$

Referring to equation 54, which gives twisting moment in terms of horse-power, and putting the two expressions equal to each other, we have :

$$T = \frac{63{,}025H}{N} = \mu PR;$$

or,

$$H = \frac{\mu NPR}{63{,}025}. \qquad (90)$$

This expression gives at once the horse-power that the clutch will transmit with a given end thrust P.

In Fig. 49 the equilibrium of the forces is shown in the little sketch at the left of the figure. The clutch faces are supposed to be in gear, and the extra force necessary to slide the two together is not considered, as it is of small importance. The static equations then are :

$$P = 2\frac{V}{2}\sin a;$$

or,

$$V = P \operatorname{cosec} a. \qquad (91)$$

$$W = \mu V = \mu P \operatorname{cosec} a. \qquad (92)$$

$$T = WR = \mu PR \operatorname{cosec} a. \qquad (93)$$

$$T = \frac{63{,}025H}{N} = \mu PR \operatorname{cosec} a;$$

or.

$$H = \frac{\mu NPR \operatorname{cosec} a}{63{,}025}. \qquad (94)$$

In Fig. 50, P would of course be variable, depending on the inclination of the little link. The amount of horse-power which this clutch would transmit would be the same as in the case of the device illustrated in Fig. 49, for an equal normal force V produced.

The further theoretical design of such clutches should be in accordance with the same principles as for arms and webs of pulleys, gears, etc. The length of the hubs must be liberal in order to prevent tipping on the shaft as a result of uneven wear. The end thrust is apt to be considerable; and extra side stiffness must be provided, as well as a rim that will not spring under the radial pressure.

PRACTICAL MODIFICATION. It is desirable to make the most complicated part of a friction clutch the driven part, for then the mechanism requiring the closest attention and adjustment may be brought to and kept at rest when no transmission of power is desired.

Simplicity is an important practical requirement in clutches. The wearing surfaces are subjected to severe usage; and it is essential that they be made not only strong in the first place, but also capable of being readily replaced when worn out, as they are sure to be after some service.

The form of clutch shown in Fig. 50 is the most efficient form of the three shown, although its commercial design is considerably different from that indicated. Usually the jaws grip both sides of the rim, pinching it between them. This relieves the clutch rim of the radial unbalanced thrust.

Adjusting screws must be provided for taking up the wear, and lock nuts for maintaining their position.

Theoretically, the rubbing surfaces should be of those materials whose coefficient of friction is the highest; but the practical question of wear comes in, and hence we usually find both surfaces of metal, cast iron being most common. For metal on metal the coefficient of friction μ cannot be safely assumed at more than 15 per cent, because the surfaces are sure to get oily.

A leather facing on one of the surfaces gives good results as to coefficient of friction, μ having a value, even for oily leather, of 20 per cent. Much slipping, however, is apt to burn the leather; and this is most likely to occur at high speeds.

Wood on cast iron gives a little higher coefficient of friction for an oily surface than metal on metal. Wood blocks can be so set into the face of the jaws as to be readily replaced when worn, and in such case make an excellent facing.

The angle a of a cone friction clutch of the type shown in Fig. 49, may evidently be made so small that the two parts will wedge together tightly with a very slight pressure P; or it may

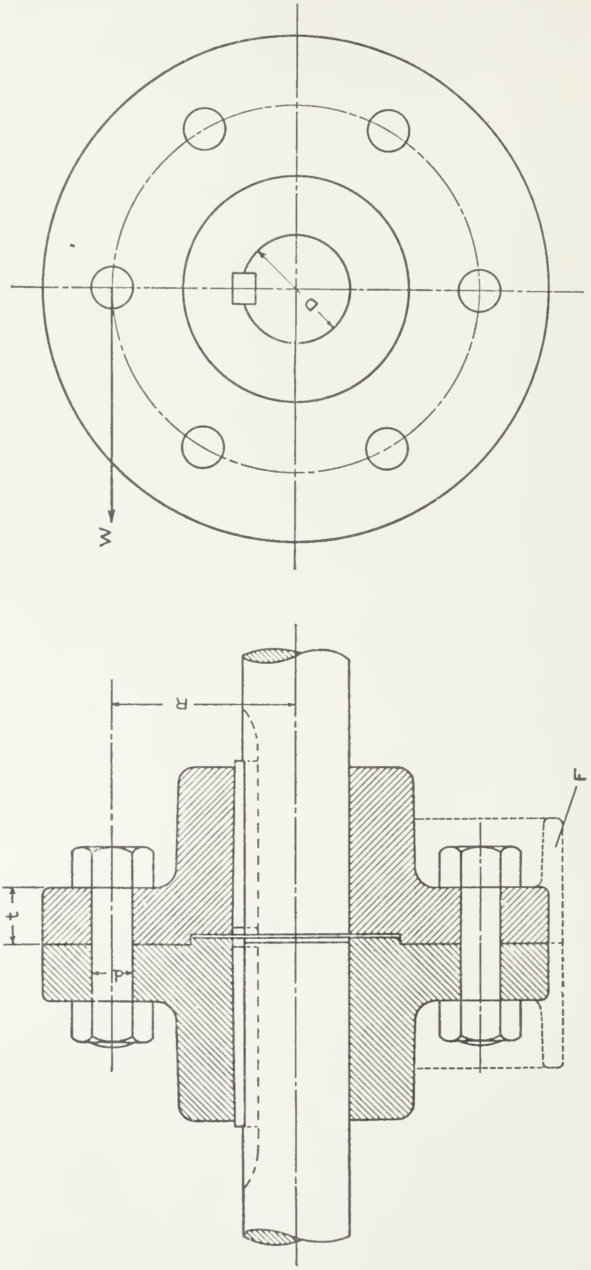


Fig. 51.

be so large as to have little wedging action, and approach the condition illustrated in Fig. 48. Between these limits there is a practical value which neither gives a wedging action so great as to make the surfaces difficult to pull apart, nor, on the other hand, requires an objectionable end thrust along the shaft in order to make the clutch drive properly.

For α = about 15°, the surfaces will free themselves when P is relieved.
" α = " 12°, " " require slight pull to be freed.
" α = " 10°, " " cannot be freed by direct pull of the hand, but require some leverage to produce the necessary force P.

PROBLEMS ON FRICTION CLUTCHES.

1. With what force must we hold a friction clutch in to transmit 30 horse-power at 200 revolutions per minute, assuming working radius of clutch to be 12 inches; coefficient of friction 15 per cent; angle $\alpha = 10°$?

2. How much horse-power could be transmitted, other conditions remaining the same, if the working radius were increased to 18 inches ?

3. What force would be necessary in problem 1, if the angle α were 15°, other conditions remaining the same ?

COUPLINGS.

NOTATION.—The following notation is used throughout the chapter on Couplings:

D =Diameter of shaft (inches).
d =Diameter of bolt body (inches).
n =Number of bolts.
R =Radius of bolt circle (inches).
S =Safe shearing fiber stress (lbs. per sq. in.).
S^c =Safe crushing fiber stress (lbs. per sq. in.).
T =Twisting moment (inch-lbs.).
t =Thickness of flange (inches).
W =Load on bolts (lbs.).

ANALYSIS. Rigid couplings are intended to make the shafts which they connect act as a solid, continuous shaft. In order that the shaft may be worked up to its full strength capacity, the coupling must be as strong in all respects as the shaft, or, in other words, it must transmit the same torsional moment. In the analysis of the forces which come upon these couplings, it is not considered that they are to take any side load, but that they are to act purely as torsional elements. It is doubtless true that in many cases they do have to provide some side strength and stiffness, but this is not their natural function, nor the one upon which their design is based.

Referring to Fig. 51, which is the type most convenient for analysis, we have an example of the simplest form of **flange coupling.** It consists merely of hubs keyed to the two portions, with flanges driving through shear on a series of bolts arranged concentrically about the shaft. The hubs, keys, and flanges are subject to the same conditions of design as the hubs, keys, and web of a gear or pulley, the key tending to shear and be crushed in the hub and shaft, and the hub tending to be torn or sheared from the flange. The driving bolts, which must be carefully fitted in reamed holes, are subject to a purely shearing stress over their full area at the joint, and at the same time tend to crush the metal in the flange, against which they bear, over their projected area. This latter stress is seldom of importance, the thickness of the flange, for practical reasons, being sufficient to make the crushing stress very low.

THEORY. The theory of hubs, keys, and flanges, being like that already given for pulleys and gears, need not be repeated for couplings. The shearing stress on the bolts is the only new point to be studied.

In Fig. 51, for a twisting moment on the shaft of T, the load at the bolt circle is $W = \frac{T}{R}$. If the number of bolts be n, equating the external force to the internal strength, we have:

$$W = \frac{T}{R} = \frac{S\pi d^2}{4} n. \qquad (95)$$

Although the crushing will seldom be of importance, yet for the sake of completeness its equation is given, thus:

$$W = \frac{T}{R} = S_c dtn. \qquad (96)$$

The internal moment of resistance of the shaft is $\frac{SD^3}{5.1}$; hence the equation representing full equality of strength between the shaft and the coupling, depending upon the shearing strength of the bolts, is:

$$\frac{SD^3}{5.1R} = \frac{S\pi d^2}{4} n. \qquad (97)$$

The theory of the other types of couplings is obscure, except as regards the proportions of the key, which are the same in all cases. The shell of the clamp coupling, Fig. 52, should be thick enough to give equal torsional strength with the shaft; but the exact function which the bolts perform is difficult to determine. In general the bolts clamp the coupling tightly on the shaft and provide rigidity, but the key does the principal amount of the driving. The bolt sizes, in these couplings, are based on judgment and relation to surrounding parts, rather than on theory.

PRACTICAL MODIFICATION. All couplings must be made with care and nicely fitted, for their tendency, otherwise, is to spring the shafts out of line. In the case of the flange coupling, the two halves may be keyed in place on the shafts, the latter then swung on centers in the lathe, and the joint faced off. Thus the joint will be true to the axis of the shaft; and, when it is clamped in position by the bolts, no springing out of line can take place.

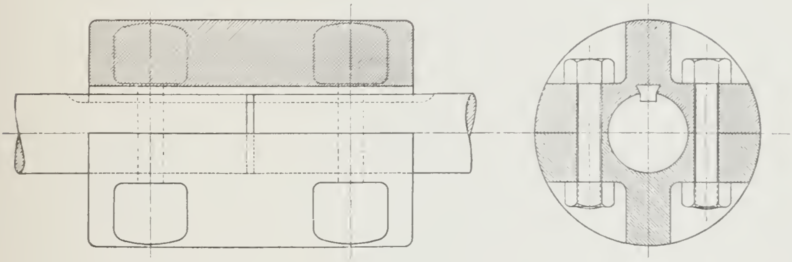


Fig. 52.

A flange F (see Fig. 51) is sometimes made on this form of coupling, in order to guard the bolts. It may be used, also, to take a light belt for driving machinery; but a side load is thereby thrown on the shaft at the joint, which is at the very point where it is desirable to avoid it.

The simplest form of rigid coupling consists of a plain sleeve slipped over from one shaft to the other, when the second is butted up against the first. This is known as a **muff coupling.** When once in place, this is a very excellent coupling, as it is perfectly smooth on the outside, and consists of the fewest possible parts, merely a sleeve and a key. It is, however, expensive to fit, difficult to remove, and requires an extra space of half its length on the shaft over which to be slipped back.

The **clamp coupling** is a good coupling for moderate-sized shafts, where the flange type of Fig. 51 would be unnecessarily expensive. The clamp coupling, Fig. 52, is simply a muff coupling split in halves, and recessed for bolts. It is cheap and is easily applied and removed, even with a crowded shaft. If bored with a piece of paper in the joint, when it is clamped in position it will pinch the shaft tightly and make a rigid connection. It is desirable to have the bolt-heads protected as much as possible, and this may be accomplished by making the outside diameter large enough so that the bolts will not project. Often an additional shell is provided to encase the coupling completely after it is located.

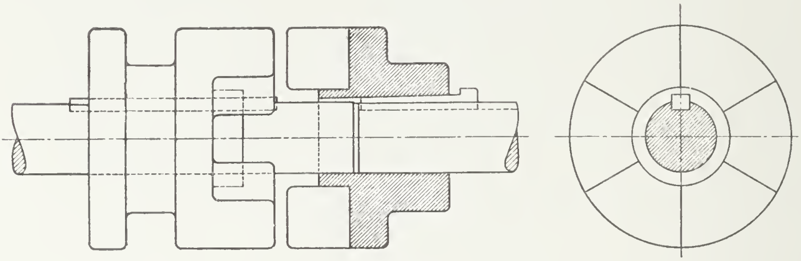


Fig. 53.

There are many other special forms of couplings, some of them adjustable. Most of them depend upon a wedging action exerted by taper cones, screws, or keys. Trade catalogues are to be sought for their description.

The **claw coupling**, Fig. 53, is nothing but a heavy flange coupling with interlocking claws or jaws on the faces of the flanges, to take the place of the driving bolts. This coupling can be thrown in or out as desired, although it usually performs the service of a rigid coupling, as it is not suited to clutching-in during rapid motion, like a friction clutch.

Flexible couplings, which allow slight lack of alignment, are made by introducing between the flanges of a coupling a flexible disc, the one flange being fastened to the inner circle of the disc, the other to the outer circle. This is also accomplished by providing the faces of the flange coupling with pins that drive by pressure together or through leather straps wrapped round the pins. These devices are mostly of a special and often uncertain nature, lacking the positiveness which is one essential feature of a good coupling.

PROBLEMS ON COUPLINGS.

1. A flange coupling of the type of Fig. 51 is used on a shaft 2 inches in diameter. The hub is 3 inches long, and carries a standard key, of proportions indicated below in the table of "Proportions for Gib Keys" (page 166). The bolt circle is 7 inches in diameter, and it is desired to use $\frac{5}{8}$-inch bolts. How many bolts are needed to transmit 60,000 inch-lbs., for a fiber stress in the bolt of 6,000 ?

2. Using 6 bolts, what diameter of bolt would be required ?

3. If four $\frac{3}{4}$-inch bolts were used on a circle of 8 inches diameter, what diameter of shaft would be used in the coupling to give equal strength with the bolts ?

BOLTS, STUDS, NUTS, AND SCREWS.

NOTATION—The following notation is used throughout the chapter on Bolts, Studs, Nuts, and Screws:

d = Diameter of bolt (inches).
d_1 = Diameter at root of thread (inches).
H = Height of nut (inches).
I = Initial axial tension (lbs.).
k = Allowable bearing pressure on surface of thread (lbs. per sq. in.).
L = Lead, or distance nut advances along axis in one revolution (inches).
l = Length of wrench handle (inches)
n = Number of threads in nut $= \frac{H}{p}$.
P = Axial load (lbs.).
p = Pitch of thread, or distance between similar points on adjacent threads, measured parallel to axis (inches).
S = Fiber stress (lbs. per sq. in.).
W = Load on bolt (lbs.).

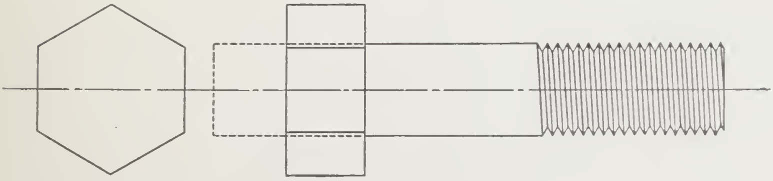


Fig. 54.

ANALYSIS. A **bolt** is simply a cylindrical bar of metal upset at one end to form a head, and having a thread at the other end, Fig. 54. A **stud** is a bolt in which the head is replaced by a thread; or it is a cylindrical bar threaded at both ends, usually having a small plain portion in the middle, Fig. 55. The object of bolts and studs is to clamp machine parts together, and yet permit these same parts to be readily disconnected. The bolt passes through the pieces to be connected, and, when tightened, causes surface compression between the parts, while the reactions on the head and nut produce tension in the bolt. Studs and tap bolts pass through one of the connected parts and are screwed into the other, the stud remaining in position when the parts are disconnected.

As all materials are elastic within certain limits, the action of a bolt in clamping two machine parts together, more especially if there is an elastic packing between them, may be represented diagrammatically by Fig. 56, in which a spring has been introduced to take the compression due to screwing up the nut. Evidently the tension in the bolt is equal to the force necessary to compress the spring. Now, suppose that two weights, each equal to $\frac{1}{2}$ W, are placed symmetrically on either side of the bolt, then the tension in the bolt will be increased by the added weights if the bolt is perfectly rigid. The bolt, however, stretches; hence some of the compression on the spring is relieved and the total tension in the

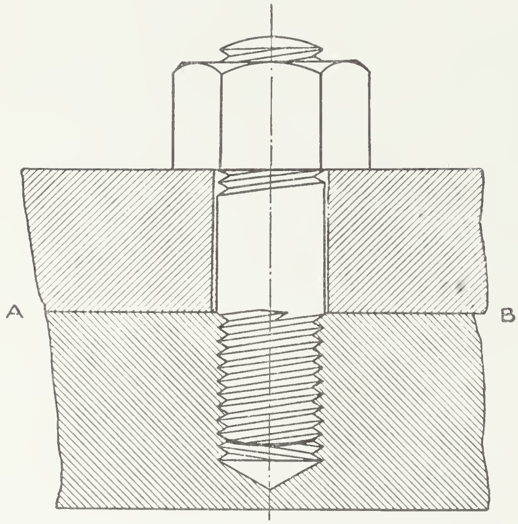


Fig. 55.

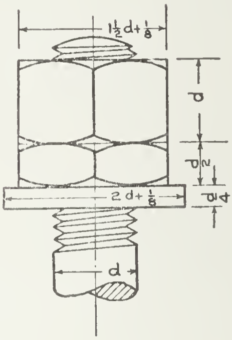


Fig. 55*a*.

bolt is less than $W + I$, by an amount depending on the relative elasticity of the bolt and spring.

Suppose that the stud in Fig. 55 is one of the studs connecting the cover to the cylinder of a steam engine, and that the studs have a small initial tension; then the pressure of the steam loads each stud, and, if the studs stretch enough to relieve the initial pressure between the two surfaces, then their stress is due to the steam pressure only; or, from Fig. 56, when $I = W$; the initial pressure due to the elasticity of the joint is entirely relieved by the assumed stretch of the studs. Except to prevent leakage, it is seldom necessary to consider the initial tension, for the stretch of the bolt may be counted on to relieve this force, and the working tension on the bolt is simply the load applied.

For shocks or blows, as in the case of the bolts found on the marine type of connecting-rod end, the stretch of the bolts acts like a spring to reduce the resulting tensions. So important is this feature that the body of the bolt is frequently turned down to the diameter of the bottom of the thread, thus uniformly distributing the stretch through the full length of the bolt, instead of localizing it at the threaded parts.

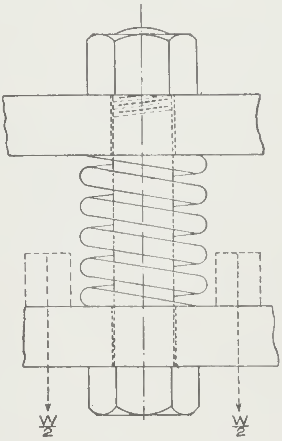


Fig. 56.

In tightening up a bolt, the friction at the surface of the thread produces a twisting moment, which increases the stress in the bolts, just as in the case of shafting under combined tension and torsion; but the increase is small in amount, and may readily be taken care of by permitting low values only for the fiber stress.

In a flange coupling, bolts are acted upon by forces perpendicular to the axis, and hence are under pure shearing stress. If the torque on the shaft becomes too great, failure will occur by the bolts shearing off at the joint of the coupling.

A bolt under tension communicates its load to the nut through the locking of the threads together. If the nut is thin, and the number of threads to take the load few, the threads may break or shear off at the root. With a V thread there is produced a component force, perpendicular to the axis of the bolt, which tends to split the nut.

In screws for continuous transmission of motion and power, the thread may be compared to a rough inclined plane, on which a small block, the nut, is being pushed upward by a force parallel to the base of the plane. The angle at the bottom of the plane is the angle of the helix, or an angle whose tangent is the lead divided by the circumference of the screw. The horizontal force corresponds to the tangential force on the screw. The friction at the surface of the thread produces a twisting moment about the axis of the screw, which, combined with the axial load, subjects the screw to combined tension and torsion. Screws with square threads are generally used for this service, the sides of the thread exerting no bursting pressure on the nut. The proportions of screw thread for transmission of power depend more on the bearing pressure than on strength. If the bearing surface be too small and lubrication poor, the screw will cut and wear rapidly.

THEORY. A direct tensile stress is induced in a bolt when it carries a load exerted along its axis. This load must be taken by the section of the bolt at the bottom of the thread. If the area at the root of the thread is $\frac{\pi d_1^2}{4}$, and if S is the allowable stress per square inch, then the internal resistance of the bolt is $\frac{S\pi d_1^2}{4}$. Equating the external load to the internal strength we have:

$$W = \frac{S\pi d_1^2}{4}. \qquad (98)$$

For bolts which are used to clamp two machine parts together so that they will not separate under the action of an applied load, the initial tension of the bolt must be at least equal to the applied load. If the applied load is W, then the parts are just about to separate when $I = W$. Therefore the above relation for strength is applicable. As the initial tension to prevent separation should be a little greater than W, a value of S should be chosen so that there will be a margin of safety. For ordinary wrought iron and steel, S may be taken at 6,000 to 8,000.

If, however, the joints must be such that there is no leakage between the surfaces, as in the case of a steam cylinder head, and supposing that elastic packings are placed in the joints, then a much larger margin should be made, for the maximum load which may come on the bolt is I + W, where W is the proportional share of the internal pressure carried by the bolt. In such cases $S = 3,000$ to $5,000$, using the lower value for bolts of less than $\frac{3}{4}$-inch diameter.

The table given on page 154 will be found very useful in proportioning bolts with U. S. standard thread for any desired fiber stress.

To find the initial tension due to screwing up the nut, we may assume the length of the handle of an ordinary wrench, measured from the center of the bolt, as about 16 times the diameter of the bolt. For one turn of the wrench a force F at the handle would pass over a distance $2\pi l$, and the work done is equal to the product of the force and space, or $F \times 2\pi l$. At the same time the axial load P would be moved a distance p along the axis. Assuming that there is no friction, the equation for the equality of the work at the handle and at the screw is:

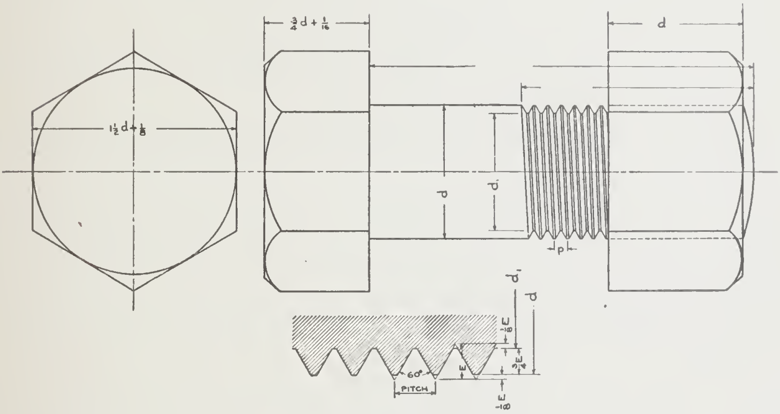


Fig. 56*a*.

$$F 2\pi l = P p. \qquad (99)$$

Friction, however, is always present; hence the ratio of the useful work (Pp) to the work applied ($F 2\pi l$) is not unity as above re-

TABLE FOR STRENGTH OF BOLTS.—U. S. Standard Thread.

BOLT.		DIAMETERS.		AREAS.		APPROXIMATE TENSILE STRENGTH. AT BOTTOM OF THREAD. (IN HUNDREDS OF POUNDS.)					APPROXIMATE SHEARING STRENGTH. FULL BOLT DIAMETER. (IN HUNDREDS OF POUNDS.)			
Diameter. Inches.	Threads per inch.	Bottom of Thread.	Tap Drill.	Bolt Body.	Bottom of Thread.	At 4,000 lbs. per sq. in.	At 5,000 lbs. per sq. in.	At 6,000 lbs. per sq. in.	At 7,000 lbs. per sq. in.	At 10,000 lbs. per. sq. in.	At 4,000 lbs. per sq. in.	At 5,000 lbs. per sq. in.	At 6,000 lbs. per sq. in.	At 7,000 lbs. per sq. in.
¼	20	.18	3/16	.05	.03	1.08	1.35	1.60	1.88	2.69	1.96	2.45	2.95	3.43
5/16	18	.24	¼	.08	.04	1.82	2.27	2.72	3.18	4.54	3.04	3.83	4.60	5.37
⅜	16	.29	5/16	.11	.07	2.71	3.39	4.07	4.75	6.78	4.40	5.52	6.62	7.73
7/16	14	.34	23/64	.15	.09	3.73	4.67	5.60	6.53	9.33	6.00	7.51	9.02	11.00
½	13	.40	13/32	.20	.13	5.00	6.25	7.50	8.75	12.00	7.84	9.81	12.00	14
9/16	12	.45	15/32	.25	.16	6.48	8.10	9.62	11.00	16	9.92	12.00	15	17
⅝	11	.51	17/32	.31	.20	8.04	10.00	12.00	14	20	12.00	15	18	21
¾	10	.62	⅝	.44	.30	12.00	15	18	21	30	18	22	26	31
⅞	9	.73	¾	.60	.42	17	24	25	29	42	24	30	36	42
1	8	.84	27/32	.78	.55	22	27	33	38	55	31	39	47	55
1⅛	7	.94	31/32	.99	.69	28	34	41	48	69	40	50	60	70
1¼	7	1.06	1 3/32	1.23	.89	31	39	47	55	78	49	61	74	86
1⅜	6	1.16	1 3/16	1.48	1.05	42	53	64	72	106	59	74	89	104
1½	6	1.28	1 9/32	1.77	1.29	51	64	77	90	128	71	88	106	124
1⅝	5½	1.39	1 13/32	2.07	1.51	61	78	92	109	153	83	104	124	145
1¾	5	1.49	1½	2.40	1.74	70	88	106	123	176	96	120	144	168
1⅞	5	1.61	1⅝	2.76	2.05	81	101	122	142	203	110	138	166	193
2	4½	1.71	1¾	3.14	2.30	92	115	138	161	230	126	157	186	220
2¼	4½	1.96	1 31/32	3.98	3.02	125	156	187	218	312	159	199	238	278
2½	4	2.17	2 3/16	4.91	3.72	148	185	222	259	370	196	245	294	344
2¾	4	2.42	2 7/16	5.94	4.62	184	230	276	322	460	237	297	356	416
3	3½	2.63	2⅝	7.07	5.43	218	272	326	381	544	283	353	424	495
3¼	3½	2.88	2 29/32	8.29	6.51	264	330	396	462	660	332	415	498	581
3½	3¼	3.10	3 1/32	9.62	7 55	302	376	452	528	754	385	481	577	673
3¾	3	3.32	3 11/32	11.04	8.64	344	430	516	602	860	442	552	663	773
4	3	3.57	3 19/32	12.57	9.99	396	495	594	693	990	503	628	754	880
4¼	2⅞	3.80	3 13/16	14.19	11.33	452	565	678	791	1,130	567	709	851	993
4½	2¾	4.03	4 1/32	15.90	12.74	507	634	760	888	1,268	636	795	954	1,113
4¾	2⅝	4.25	4 9/32	17.72	14.22	567	709	851	1,993	1,420	709	886	1,063	1,240
5	2½	4.48	4½	19.63	15.76	630	788	946	1,103	1,676	785	982	1,178	1,374
5½	2⅜	4.95	4 31/32	23.76	19.27	770	962	1,154	1,347	1,924	950	1,188	1,425	1,663
6	2¼	5.42	5 7/16	28.27	23.09	923	1,159	1,384	1,617	2,307	1,128	1,414	1,696	1,979

lations assume. From numerous experiments on the friction of screws and nuts, it has been found that the efficiency may be as low as 10 per cent. Introducing the efficiency in above equation, it may be written:

$$\frac{Pp}{F2\pi l} = \frac{1}{10}. \qquad \textbf{(100)}$$

Assuming that 50 pounds is exerted by a workman in tightening up the nut on a 1-inch bolt, the equation above shows that P = 4,021 pounds; or the initial tension is somewhat less than the tabular safe load shown for a 1-inch bolt, with S assumed at 10,000 pounds per sq. inch.

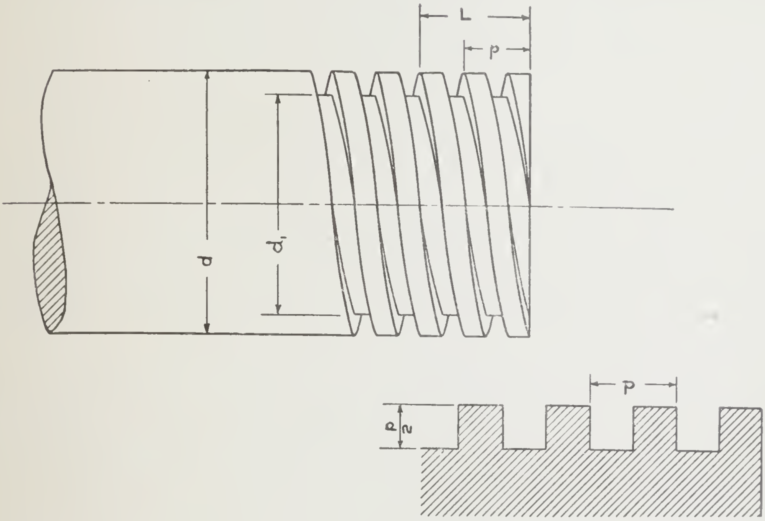


Fig. 58.

For shearing stresses the bolt should be fitted so that the body of the bolt, not the threads, resists the force tending to shear off the bolt perpendicular to its axis. The internal strength of the bolt to resist shear is the allowable stress S times the area of the bolt in shear, or $\frac{S\pi d^2}{4}$. If W represents the external force tending to shear the bolt the equality of the external force to the internal strength is:

$$W = \frac{S\pi d^2}{4}. \qquad \textbf{(101)}$$

Reference to the table on page 154 for the shearing strength of bolts, may be made to save the labor of calculations.

Let Fig. 58 represent a square thread screw for the transmission of motion. The surface on which the axial pressure bears, if n is the number of threads in the nut, is $\frac{\pi}{4}(d^2 - d_1^2)\,n$. Suppose that a pressure of k pounds per square inch is allowed on the surface of the thread. Then the greatest permissible axial load P must not exceed the allowable pressure; or, equating,

$$P = k\,\frac{\pi}{4}(d^2 - d_1^2)\,n. \qquad \text{(102)}$$

The value of k varies with the service required. If the motion be slow and the lubrication very good, k may be as high as 900. For rapid motion and doubtful lubrication, k may not be over 200. Between these two extremes the designer must use his judgment, remembering that the higher the speed the lower is the allowable bearing pressure.

PRACTICAL MODIFICATION. It will be noted in the formulæ for bolt strengths that different values for S are assumed. This is necessary on account of the uncertain initial stresses which are produced in setting up the nuts. For cases of mere fastening, the safe tension is high, as just before the joint opens the tension is about equal to the load and yet the fastening is secure. On the other hand, bolts or studs fastening joints subjected to internal fluid pressure must be stressed initially to a greater amount than the working pressure which is to come on the bolt. As this initial stress is a matter of judgment on the part of the workman, the designer, in order to be on the safe side, should specify not less than $\frac{5}{8}$-inch or $\frac{3}{4}$-inch bolts for ordinary work, so that the bolts may not be broken off by a careless workman accidentally putting a greater force than necessary on the wrench handle. In making a steam-tight joint, the spacing of the bolts will generally determine their number; hence we often find an excess of bolt strength in joints of this character.

Through bolts are preferred to studs, and studs to tap bolts or cap screws. If possible, the design should be such that through bolts may be used. They are cheapest, are always in standard stock, and well resist rough usage in connecting and disconnecting. The threads in cast iron are weak and have a tendency to crumble; and if a through bolt cannot be used in such a case, a stud, which can be placed in position once for all, should be employed—not a tap bolt, which injures the thread in the casting every time it is removed.

The plain portion of a stud should be screwed up tight against the shoulder, and the tapped hole should be deep enough to prevent bottoming. To avoid breaking off the stud at the shoulder, a neck, or groove, may be made at the lower end of the thread entering the nut.

To withstand shearing forces the bolts must be fitted so that no lost motion may occur, otherwise pure shearing will not be secured.

Nuts are generally made hexagonal, but for rough work are often made square. The hexagonal nut allows the wrench to turn through a smaller angle in tightening up, and is preferred to the square nut. Experiments and calculations show that the height of the nut with standard threads may be about $\frac{1}{2}$ the diameter of the bolt and still have the shearing strength of the thread equal to the tensile strength of the bolt at the root of the thread. Practically, however, it is difficult to apply such a thin wrench as this proportion would call for on ordinary bolts. More commonly the height of the nut is made equal to the diameter of the bolt so that the length of thread will guide the nut on the bolt, give a low bearing pressure on the threads, and enable a suitable wrench to be easily applied. The standard proportions for bolts and nuts may be found in any handbook. Not all manufacturers conform to the United States standard; nor do manufacturers in all cases conform to one another in practice.

If the bolt is subject to vibration, the nuts have a tendency to loosen. A common method of preventing this is to use double nuts, or **lock nuts**, as they are called (see Fig. 55 A). The under nut is screwed tightly against the surface, and held by a wrench while the second nut is screwed down tightly against the first. The effect is to cause the threads of the upper nut to bear against the under sides of the threads of the bolt. The load on the bolt is sustained therefore by the upper nut, which should be the thicker of the two ; but for convenience in applying wrenches the position of the nuts is often reversed.

The form of thread adapted to transmitting power is the square thread, which, although giving less bursting pressure on the nut, is not as strong as the V thread for a given length, since the total section of thread at the bottom is only $\frac{1}{2}$ as great. If the pressure is to be transmitted in but one direction, the two types may be combined advantageously to form the buttress thread of the proportions shown in Fig. 59. Often, as in the carriage of a lathe, to allow the split nut to be opened and closed over the lead screw, the sides of the thread are placed at a small angle, say 15°, to each other, as illustrated in Fig. 60.

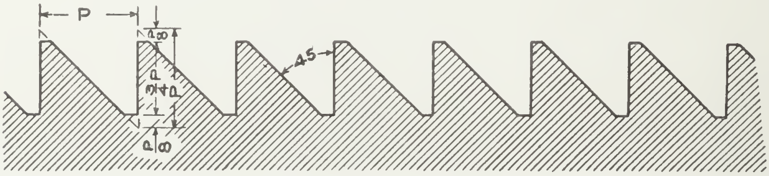


Fig. 59.

The practical commercial forms in which we find screwed fastenings are included in five classes, as follows:

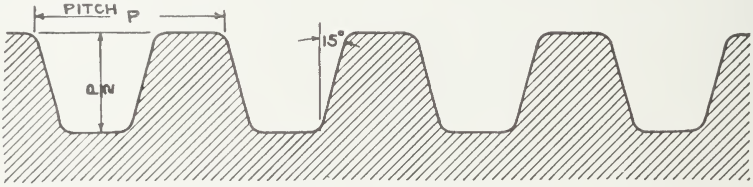


Fig. 60.

1. **Through bolts** (Fig. 61), usually rough stock, with square upset heads, and square or hexagonal nuts.

2. **Tap bolts** (Fig. 62), also called **cap screws.** These usually have hexagonal heads, and are found both in the rough form, and finished from the rolled hexagonal bar in the screw machine.

3. **Studs** (Fig. 63), rough or finished stock, threaded in the screw machine.

4. **Set screws** (Fig. 64), usually with square heads, and case-hardened points. Many varieties of set screws are made, the

principal distinguishing feature of each being in the shape of the point. Thus, in addition to the plain beveled point, we find the "cupped," rounded, conical, and "teat" points.

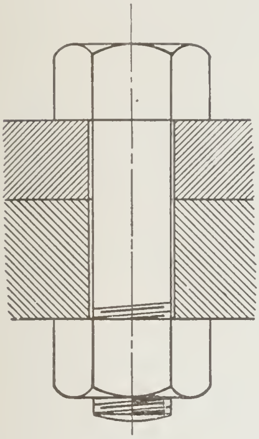


Fig. 61.

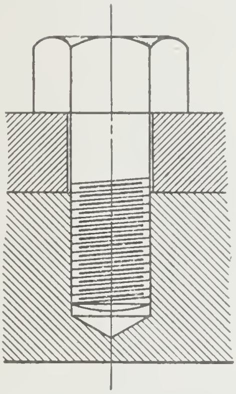


Fig. 62.

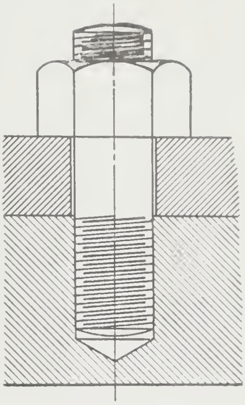


Fig. 63.

5. **Machine screws** (Fig. 64*a*), usually round, "button," or countersunk head. Common proportions are indicated relative to diameter of body of screw.

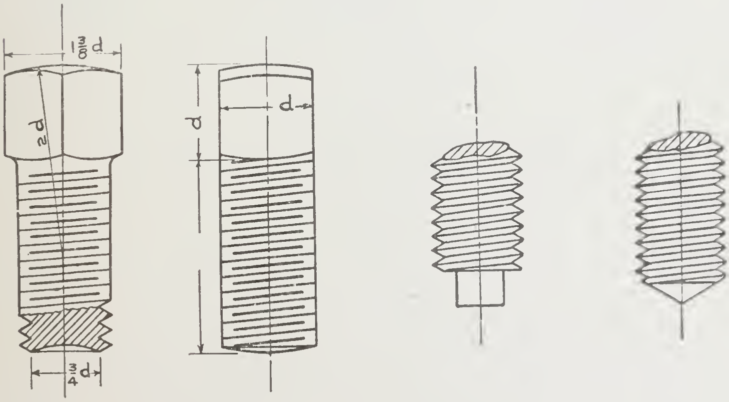


Fig. 64.

PROBLEMS ON BOLTS, STUDS, NUTS, AND SCREWS.

1. Calculate the diameter of a bolt to sustain a load of 5,000 lbs.

2. The shearing force to be resisted by each of the bolts of a flange coupling is 1,200 lbs. What commercial size of bolt is required?

3. With a wrench 16 times the diameter of the bolt, and an efficiency of 10 per cent, what axial load can a man exert on a standard $\frac{3}{4}$-inch bolt, if he pulls 40 lbs. at the end of the wrench handle?

4. A single, square-threaded screw of diameter 2 inches, lead $\frac{1}{4}$ inch, depth of thread $\frac{1}{8}$ inch, length of nut 3 inches, is to be allowed a bearing pressure of 300 lbs. per square inch. What axial load can be carried?

5. Calculate the shearing stress at the root of the thread in problem 4.

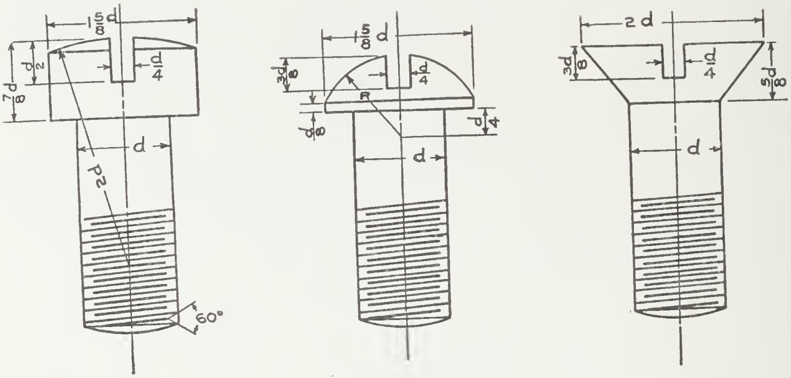


Fig. 64*a*.

KEYS, PINS, AND COTTERS.

NOTATION—The following notation is used throughout the chapter on Keys, Pins, and Cotters:

D = Average diameter of rod (inches).
D_1 = Outside diameter of socket (inches).
d = Diameter of shaft (inches).
L = Length of key (inches).
P = Driving force (lbs.).
P_1 = Axial load on rod (lbs.).
R = Radius at which P acts (inches).
S_c = Safe crushing fiber stress (lbs. per sq. in.).
S_s = Safe shearing fiber stress (lbs. per sq. in.).
S_t = Safe tensile fiber stress (lbs. per sq. in.).
T = Thickness of key (inches).
W = Width of key (inches).
w = Average width of cotter (inches).
w_1 = End of slot to end of rod (inches).
w_2 = End of slot to end of socket (inches).

KEYS AND PINS.

ANALYSIS. Keys and pins are used to prevent relative rotary motion between machine parts intended to act together as one piece. If we drill completely through a hub and across the shaft, and insert a tightly fitted pin, any rotary motion of the one will be transmitted to the other, provided the pin does not fail by shearing off at the joint between the shaft and the hub. The shearing area is the sum of the cross-sections of the pin at the joint.

We may drill a hole in the joint, the axis of the hole being parallel to the axis of the shaft, and drive in a pin, in which case we introduce a shearing area as before, but the area is now equal to the diameter of the pin multiplied by its length, and the pin is stressed sidewise, instead of across. It is evident in the sidewise case that we may increase the shearing area to anything we please, without changing the diameter of the pin, merely by increasing the length of the pin.

As there are some manufacturing reasons why a round pin placed lengthwise in the joint is not always applicable, we may make the pin a rectangular one, in which case it is called a **key.**

When pins are driven across the shaft as in the first instance, they are usually made taper. This is because it is easier to ream a taper hole to size than a straight hole, and a taper pin will drive more easily than a straight pin, it not being necessary to match the hole in hub and shaft so exactly in order that the pin may enter. The taper pin will draw the holes into line as it is driven, and can be backed out readily in removal.

Keys of the rectangular form are either straight or tapered, but for different reasons from those just stated for pins. Straight keys have working bearing only at the sides, driving purely by shear, crushing being exerted by the side of the key in both shaft and hub, over the area against the key. The key itself does not prevent end motion along the shaft; and if end motion is not desired, auxiliary means of some sort must be resorted to, as, for example, set screws through the hub jamming hard against the top of the key.

If end motion along the shaft is desired, the key is called a **spline**, and, while not jammed against the shaft, is yet prevented from changing its relation to the hub by some means such as illustrated in Fig. 65.

Taper keys not only drive through sidewise shearing strength, but prevent endwise motion by the wedging action exerted between the shaft and hub. These keys drive more like a strut from corner to corner; but this action is incidental rather than intentional, and the proportions of a taper key should be such that it will give its full resisting area in shearing and crushing, the same as a straight key.

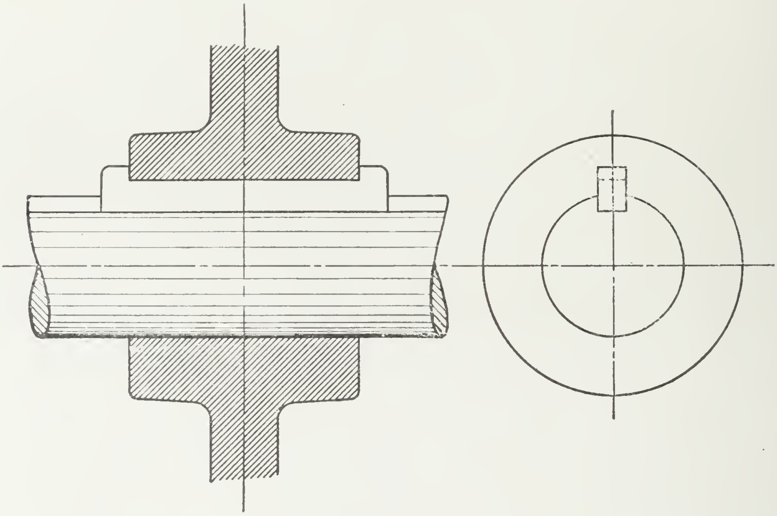


Fig. 65.

THEORY. Suppose that the pin illustrated in Fig. 66 passes through hub and shaft, and the driving force P acts at the radius R; then the force which is exerted at the surface of the shaft to shear off the pin at the points A and B is $\frac{2\,PR}{d}$. If D_1 is the average diameter of the pin, its shearing strength is $\frac{2\pi D_1^2 S_s}{4}$. Equating the external force to the internal strength, we have:

$$\frac{2PR}{d} = \frac{2\pi D_1^2\, S_s}{4};$$

or,

$$D_1 = \sqrt{\frac{4PR}{\pi d S_s}}. \qquad (103)$$

In Fig. 67 a rectangular key is sunk half way in hub and shaft according to usual practice. Here the force at the surface of the shaft, calculated the same as before, not only tends to shear off the key along the line AB, but tends to crush both the portion in the shaft and in the hub. The shearing strength along the

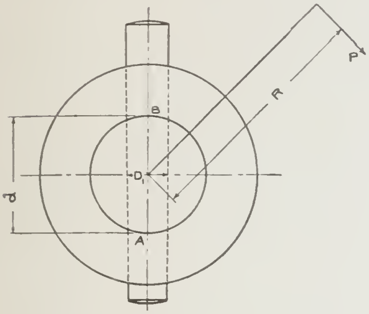


Fig. 66.

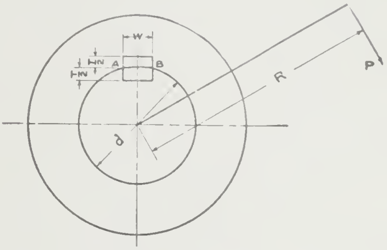


Fig. 67.

line AB is LWS_s. Equating external force to internal strength, we have:

$$\frac{2PR}{d} = LWS_s\,;$$

or,

$$W = \frac{2PR}{dLS_s}. \qquad (104)$$

The crushing strength is, of course, that due to the weaker metal, whether in shaft or hub. Let S_c be this least safe crushing fiber stress. The crushing strength then is $\frac{LT}{2}S_c$, and, equating external force to internal strength, we have:

$$\frac{2PR}{d} = \frac{LT}{2}S_c\,;$$

or,

$$T = \frac{4PR}{dLS_c}. \qquad (105)$$

The proportions of the key must be such that the equations as above, both for shearing and for crushing, shall be satisfied.

PRACTICAL MODIFICATION. Pins across the shaft can be used to drive light work only, for the shearing area cannot be very large. A large pin cuts away too much area of the shaft, decreasing the latter's strength. Pins are useful in preventing end motion, but in this case are expected to take no shear, and may be of small diameter. The common split pin is especially adapted to this service, and is a standard commercial article.

Taper pins are usually listed according to the Morse standard taper, proportions of which may be found in any handbook. It is desirable to use *standard* taper pins in machine construction, as the reamers are a commercial article of accepted value, and readily obtainable in the machine-tool market.

With properly fitted keys, the shearing strength is usually the controlling element. For shafts of ordinary size, the standard proportions as given in tables like that below are safe enough without calculation, up to the limit of torsional strength of the shaft. For special cases of short hubs or heavy loads, a calculation is needed to check the size, and perhaps modify it.

Splines, also known as "feather keys," require thickness greater than regular keys, on account of the sliding at the sides. A table suggesting proportions for splines is given on page 166.

Though the spline may be either in the shaft or hub, it is the more usual thing to find the spline dovetailed (Fig. 67*a*), "gibbed," or otherwise fastened in the hub; and a long spline way made in the shaft, in which it slides.

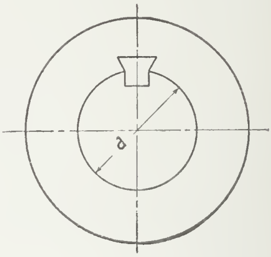


Fig. 67*a*.

The straight key, accurately fitted, is the most desirable fastening device for accurate machines, such as machine tools, on account of the fact that there is absolutely no radial force exerted to throw the parts out of true. It, however, requires a tight fit of hub to shaft, as the key cannot be relied upon to take up any looseness.

The taper key (Fig. 68), by its wedging action, will take up some looseness, but in so doing throws the parts out slightly. Or, even if the bored fit be good, if the taper key be not driven home with care, it will spring the hub, and make the parts run untrue. The great advantage, however, that the taper key has of holding the hub from endwise motion, renders it a very useful and practical article. It is usually provided with a head, or "gib," which permits a draw hook to be used to wedge between the face of the hub and the key to facilitate starting the key from its seat.

Two keys at 90° from each other may be used in cases where one key will not suffice. The fine workmanship involved in spacing these keys so that they will drive equally makes this plan inadvisable except in case of positive and unavoidable necessity.

The "Woodruff" key (Fig. 69) is a useful patented article for certain locations. This key is a half-disc, sunk in the shaft and the hub is slipped over it. A simple rotary cutter is dropped into the shaft to produce the key seat; and on account of the depth in the shaft, the tendency to rock sidewise is eliminated, and the drive is purely by shear.

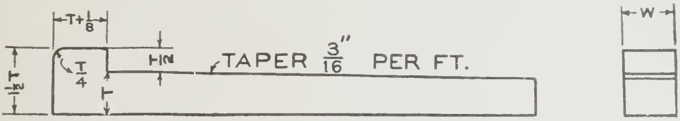


Fig. 68.

Keys may be milled out of solid stock, or drop-forged to within a small fraction of finished size. The drop-forged key is an excellent modern production and requires but a minimum amount of fitting. Any key, no matter how produced, requires some hand fitting and draw filing to bring it properly to its seat and give it full bearing.

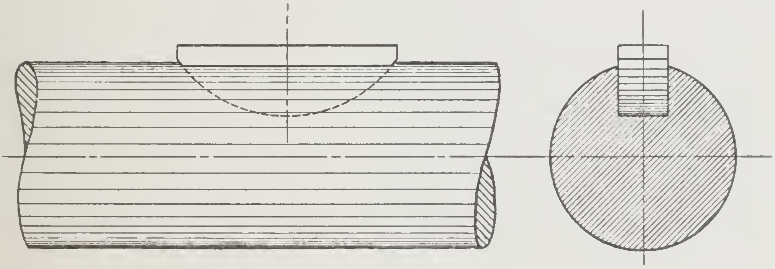


Fig. 69.

It is good mechanical policy to avoid keyed fastenings whenever possible. This does not mean that keys may never be used, but that a key is not an ideal way to produce an absolutely positive drive, partly because it is an expensive device, and partly because the tendency of any key is to work itself loose, even if carefully fitted.

The following tables are suggested as a guide to proportions of gib keys and feather keys, and will be found useful in the absence of any manufacturer's standard list:

Fig. 70. **PROPORTIONS FOR GIB KEYS.**

Diameter of shaft (*d*), inches.	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{5}{8}$	2	$2\frac{1}{2}$	$3\frac{1}{4}$	4	5	$6\frac{1}{2}$
Width (W), inches.	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{5}{16}$	$1\frac{5}{8}$
Thickness (T), inches.	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{13}{32}$	$\frac{7}{16}$	$\frac{17}{32}$	$\frac{21}{32}$	$\frac{13}{16}$	1	$1\frac{1}{4}$

Fig. 71. **PROPORTIONS FOR FEATHER KEYS.**

Diameter of Shaft (*d*), inches,	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$
Width (W), inches.	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$
Thickness (T), inches.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$

COTTERS.

ANALYSIS. Cotters are used to fasten hubs to rods rather than shafts, the distinction between a rod and a shaft being that a rod takes its load in the direction of its length, and does not drive by rotation. A cotter, therefore, is nothing but a cross-pin of modified form, to take shearing and crushing stress in the direction of the axis of the rod, instead of perpendicular to it.

Referring to Fig. 72, one will see that the cotter is made long and thin—long, in order to get sufficient shearing area to resist shearing along lines A and B; thin, in order to cut as little cross-sectional area out of the body of the shaft as possible. The cotter itself tends to shear along the lines A and B, and crush along the surfaces K, G, and J. The socket tends to crush along the surfaces K and G. The rod end D tends to be sheared out along the lines C H and Q E, and also to be crushed along the surface J. The socket tends to be sheared along the lines V U and X Y.

The cotter is made taper on one side, thus enabling it to draw up the flange of the rod tightly against the head of the socket. This taper must not be great enough to permit easy "backing out" and loosening of the cotter under load or vibration in the rod. In responsible situations this cannot be safely guarded against except through some auxiliary locking device, such as lock nuts on the end of the cotter (Fig. 73).

THEORY. Referring to Fig. 72, assume an axial load of P_1, as shown. The successive equations of external force to internal strength are enumerated below, for the different actions that take place:

For shearing along lines A and B, w being the average width of cotter, and S_s safe shearing stress of cotter,

$$P_1 = 2TwS_s. \qquad (106)$$

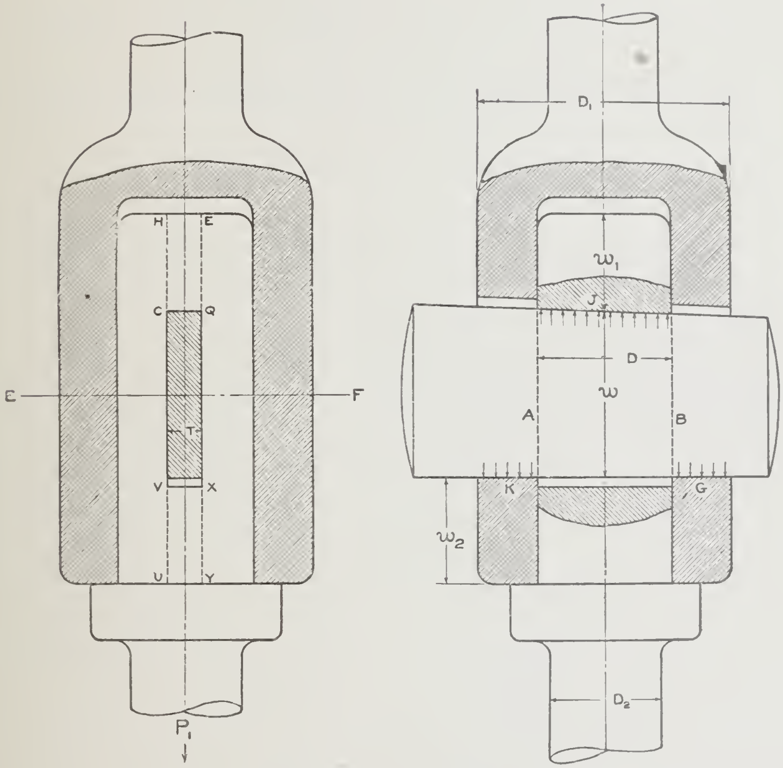


Fig. 72.

For crushing along surfaces K and G, S_c being least safe crushing stress, whether of cotter or socket,

$$P_1 = T(D_1 - D)S_c. \qquad (107)$$

For crushing along surface J, S_c being least safe crushing stress, whether of cotter or socket,

$$P_1 = DTS_c. \qquad (108)$$

For shearing along surfaces CH and QE, S_s being safe shearing stress of rod end, and w_1 end of slot to end of rod,

$$P_1 = 2w_1DS_s. \tag{109}$$

For tension in rod end at section across slot, S_t being safe tensile stress in rod end,

$$P_1 = (\frac{\pi D^2}{4} - TD)S_t. \tag{110}$$

For tension in socket at section across slot, S_t being safe tensile stress in socket,

$$P_1 = [\frac{\pi D_1^2}{4} - \frac{\pi D^2}{4} - T(D_1 - D)]S_t. \tag{111}$$

For shearing in socket along the lines VU and XY, S_s being safe shearing stress in the socket, and w_2 end of slot to end of socket,

$$P_1 = 2w_2(D_1 - D)S_s. \tag{112}$$

The proportions of cotter and socket may be fixed to some extent by practical or assumed conditions. The dimensions may then be tested by the above equations, that the safe working stresses may not be exceeded, the dimensions being then modified accordingly.

The steel of which both cotter and rod would ordinarily be made has range of working fiber stress as follows :

Tension, 8,000 to 12,000 (lbs. per sq. in.)
Compression, 10,000 to 16,000 (lbs. per sq. in.)
Shear, 6,000 to 10,000 (lbs. per sq. in.)

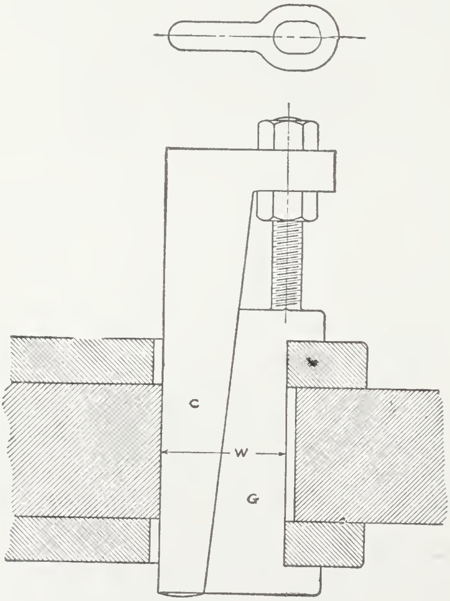


Fig. 73.

The socket, if made of cast iron, will be weak as regards tension, tendency to shear out at the end, and tendency to split. The uncertainty of cast iron to resist these is so great that the hub or socket must be very clumsy in order to have enough surplus strength. This is always a noticeable feature of the cotter type of fastening, and cannot well be avoided.

PRACTICAL MODIFICATION. The driving faces of the cotter are often made semicircular. This not only gives more shearing area at the sides of the slots, but makes the production of the slots easier in the shop. It also avoids the general objection to sharp corners—namely, a tendency to start cracks.

A practicable taper for cotters is ½ inch per foot. This will under ordinary circumstances prevent the cotter from backing out under the action of the load. When set screws against the side of the cotter, or lock nuts are used, as in Fig. 73, the taper may be greater than this, perhaps as much as 1½ inches per foot.

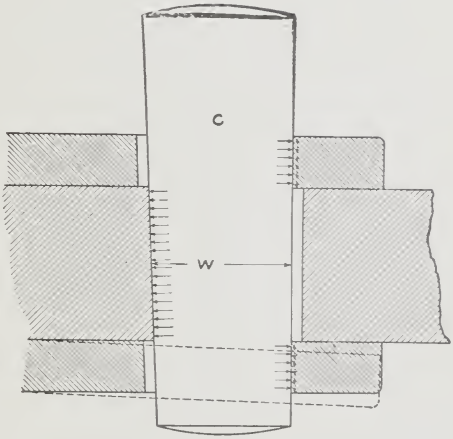


Fig. 74.

In the common use of the cotter for holding the strap at the ends of connecting rods, the strap acts like a modified form of socket. This is shown in Figs. 73 and 74. Here, in addition to holding the strap and rod together lengthwise, it may be necessary to prevent their spreading, and for this purpose an auxiliary piece G with gib ends is used. The tendency without this extra piece is shown by the dotted lines in Fig. 74.

The general mechanical fault with cottered joints is that the action of the load, especially when it constantly reverses, as in pump piston rods, always tends to work the cotter loose. Vibration also tends to produce the same effect. Once this looseness is started in the joint, the cotter loses its pure crushing and shearing action, and begins to partake of the nature of a hammer, and pounds itself and its bearing surfaces out of their true shape. Instead of a collar on the rod, we often find a taper fit of the rod in the socket; and any looseness in this case is still worse, for the rod then has end play in the socket, and by its "shucking" back and forth tends to split open the socket.

The only answer to these objections is to provide a positive locking device, and take up any looseness the instant it appears.

PROBLEMS ON KEYS, PINS, AND COTTERS.

1. Calculate the safe load in shear which can be carried on a key $\frac{1}{2}$ inch wide, $\frac{3}{8}$ inch thick, and 5 inches long. Assume $S_s = 6{,}000$.

2. Assuming the above key to be $\frac{3}{16}$ inch in hub and $\frac{3}{16}$ inch in shaft, test its proportions for crushing, at $S_c = 16{,}000$.

3. A gear 60 inches in diameter has a load of 3,000 lbs. at the pitch line. The shaft is 4 inches in diameter, in a hub, 5 inches long; and the key is a standard gib key as given in the table. Test its proportions for shearing.

4. A piston rod 2 inches in diameter carries a cotter $\frac{3}{8}$ inch thick, and has an axial load of 20,000 lbs. Calculate the average width of the cotter. $S_s = 9{,}000$.

5. Calculate fiber stress in rod in preceding problem at section through slot.

6. How far from the end of rod must the end of slot be?

7. Calculate the crushing fiber stresses on cotter, rod, and socket.

8. How far from the end of socket must the end of slot be, assuming the socket to be of steel?

BEARINGS, BRACKETS, AND STANDS.

NOTATION—The following notation is used throughout the chapters on Bearings Brackets, and Stands.

A = Area (square inches).
a = Distance between bolt centers (inches).
b = Width of bracket base (inches).
c = Distance of neutral axis from outer fiber (inches).
D = Diameter of shaft (inches).
d = Diameter of bolt body (inches).
d_1 = Diameter at root of thread (inches).
H = Horse-power.
h = Thickness of cap at center (inches).
I = Moment of inertia.
L = Length of bearing (inches).
μ = Coefficient of friction (per cent).

N = Number of revolutions per minute.
n = Number of bolts in cap.
n_1 = Number of bolts in bracket base.
P = Total pressure on bearing (lbs.).
p = Pressure per square inch of projected area (lbs).
S = Safe tensile fiber stress (lbs.).
S_s = " shearing " (lbs.).
T = Total load on bolts at top of bracket (lbs.).
t = Thickness of bracket base (inches).
x = Distance from line of action of load to any section of bracket (inches).

ANALYSIS. Machine surfaces taking weight and pressure of other parts in motion upon them are, in general, known as **bearings.** If the motion is rectilinear, the bearing is termed a **slide, guide,** or **way,** such as the cross slide of a lathe, the cross-head guide of a steam engine, or the ways of a lathe bed.

If the motion is a rotary one, like that of the spindle of a lathe, the simple word "bearing" is generally used.

In any bearing, sliding or rotary, there must be strength to carry the load, stiffness to distribute the pressure evenly over the full bearing surface, low intensity of such pressure to prevent the lubricant from being squeezed out and to minimize the wear, and sufficient radiating surface to carry away the heat generated by friction of the surfaces as fast as it is generated. Sliding bearings are of such varied nature, and exist under conditions so peculiar to each case, that a general analysis is practically impossible beyond that given in the sentence above.

Rotary bearings can be more definitely studied, as there are but two variable dimensions, diameter and length, and it is the proper relation between these two that determines a good bearing. The size of the shaft, as noted under "Shafts," is calculated by taking the bending moment at the center of the bearing, combining it with the twisting moment, and solving for the diameter consistent with the assumed fiber stress. But this size must then be tried for deflection due to the bending load, in order that the requirement for stiffness may be fulfilled. When this is accomplished, the friction at the bearing surface may still generate so much heat that the exposed surface of the bearing will not radiate it as fast as generated, in which case the bearing gets hotter and hotter, until it finally burns out the lubricant and melts the lining of the bearing, and ruin results.

The heat condition is usually the critical one, as it is very easy to make a short bearing which is strong enough and amply stiff for the load it carries, but which nevertheless is a failure as a bearing, because it has so small a radiating surface that it cannot run cool.

The side load which causes the friction and the consequent development of heat, is due to the pull of the belt in the case of pulleys, the load on the teeth of gears, the pull on cranks and levers, the weight of parts, etc. If we could exert pure torsion on shafts without any side pressure, and counteract all the weight that comes on the shaft, we should not have any trouble with the development of heat in bearings; in fact, there would theoretically be no need of bearings, as the shafts would naturally spin about their axes, and would not need support.

It can be shown, theoretically, that the radiating surface of a bearing increases relatively to the heat generated by a given side load, *only when the length of the bearing is increased.* In other words, increasing the diameter and not the length, theoretically increases the heat generated per unit of time just as much as it increases the radiating surface; hence nothing is gained, and heat accumulates in the bearing as before. This important fact is verified by the design of high-speed bearings, which, it is always noted, are very long in proportion to their diameter, thus giving relatively high radiating power.

Bearings must be rigidly fastened to the body of the machine in some way, and the immediate support is termed a **bracket, frame,** or **housing.** "Bracket" is a very general term, and applies to the supports of other machine parts besides "bearings." It is especially applicable to the more familiar types of bearing supports, and is here introduced to make the analysis complete.

The bracket must be strong enough as a beam to take the side load, the bending moment being figured at such points as are necessary to determine its outline. It may be of solid, box, or ribbed form, the latter being the most economical of material, and usually permitting the simplest pattern. The fastening of the bracket to the main body of the machine must be broad to give stability; the bolts act partly in shear to keep the bracket from sliding along its base, and partly in tension to resist its tendency to rotate about some one of its edges, due to the side pull of the belt, gear tooth, or lever load, as the case may be. The weight of the bracket itself and of the parts it sustains through the bearing, has likewise to be considered; and this acts, in conjunction with the working load on the bearing, to modify the direction and magnitude of the resultant load on the bracket and its fastening.

Stands are forms of brackets, and are subject to the same analysis. The distinction is by no means well defined, although we usually think more readily of a stand as having an upright or inverted position with reference to the ground. The ordinary "hanger" is a good example of an inverted stand; and the regular "floor stand," found on jack shafts in some power houses, is an example of the general class.

THEORY. As the method of calculation of the diameter of the shaft, as well as its deflection, has been considered under "Shafts," we may assume that the theoretical study of bearings starts on a given basis of shaft diameter D. The main problem then being one of heat control, let us first calculate the amount of heat developed in a bearing by a given side load. The force of friction acts at the circumference of the shaft, and is equal to the coefficient of friction times the normal force; or, for a given side load P, Fig. 75, the force of friction would be μP. The peripheral speed of the shaft for N revolutions per minute is $\frac{\pi DN}{12}$ feet per minute. As work is "force times distance," the work wasted in friction is then $\frac{\mu P \pi DN}{12}$ foot-pounds per minute. One horse-power being equal to 33,000 foot-pounds per minute, we have the equation,

$$H = \frac{\mu P \pi DN}{12 \times 33{,}000}. \qquad \textbf{(113)}$$

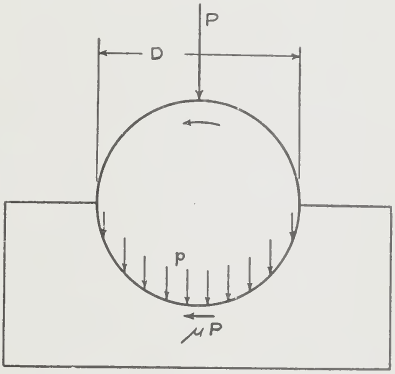


Fig. 75.

The value of μ for ordinary, well-lubricated bearings, may run as low as 5 per cent; but as the lubrication is often impaired, it quite commonly rises to 10 or 12 per cent. A value of 8 per cent is a fair average. This amount of horse-power is dissipated through the bearing in the form of heat. If we could exactly determine the ability that each particle of the metal around the shaft had to transmit the heat, or to pass it along to the outside of the casting, and if we could then determine the ability of the particles of air surrounding the casting to receive and carry away this heat, we could calculate just such proportions of the bearing and its casing as would never choke or retard this free transfer of heat away from the running surface.

Such refined theory is not practical, owing to the complicated shapes and conditions surrounding the bearing. The best that we can do is to say that for the usual proportions of bearings the side load may exist up to a certain intensity of "pressure per square inch of projected area" of bearing, or, in form of an equation,

$$P = pLD. \qquad (114)$$

The constant p is of a variable nature, depending on lubrication, speed, air contact, and other special conditions. For ordinary bearings having continuous pressure in one direction, and only fair lubrication, 400 to 500 is an average value. When the pressure changes direction at every half-revolution, the lubricant has a better chance to work fully over the bearing surface, and a higher value is permissible, say, 500 to 800. In locations where mere oscillation takes place, not continuous rotation, and reversal of pressure occurs, as on the cross-head pin of a steam engine, p may run as high as 900 to 1,200. On the crank pins of locomotives, which have the reversal of pressure, and the benefit of high velocity through the air to facilitate cooling, the pressures may run equally high. On the eccentric crank pins of punching and shearing machines, where the pressure acts only for a brief instant and at intervals, the pressure ranges still higher without any dangerous heating action.

When a bearing, for practical reasons, is provided with a cap held in place by bolts or studs, the *theory* of the cap and bolts is of little importance, unless the load comes directly against the cap and bolts. Except in the latter case, the proportions of the cap and the size of the bolts are dependent upon general appearance and utility, it being manifestly desirable to provide a substantial design, even though some excess of strength is thereby introduced.

For the worst case of loading, however, which is when the cap is acted upon by the direct load, such as P in Fig. 76, we have the condition of a centrally loaded beam supported at the bolts. It is probable that the beam is partially fixed at the ends by the clamping of the nut; also that the load P, instead of being concentrated at the center, is to some extent distributed. It is hardly fair to assume the external moment equal to $\frac{Pa}{8}$ or $\frac{Pa}{4}$, the one being too small, perhaps, and the other too large. It will be reasonable to take the external moment at $\frac{Pa}{6}$, in which case, equating the external moment to the internal moment of resistance,

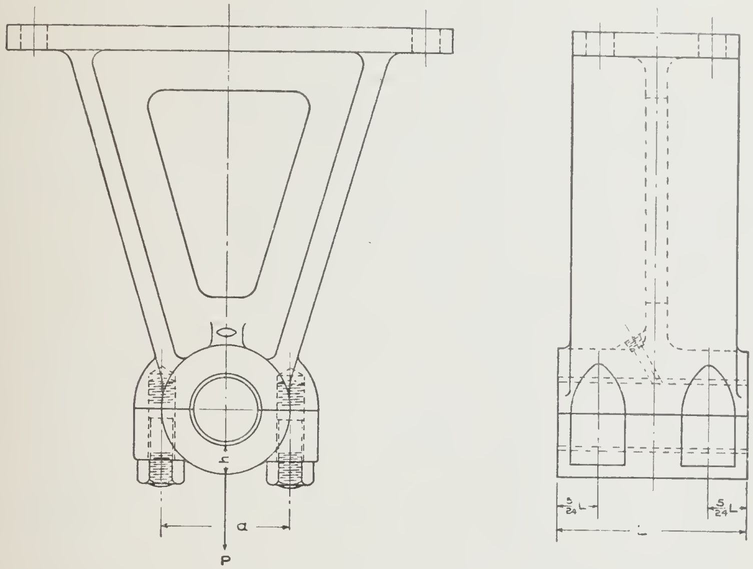


Fig. 76.

$$\frac{Pa}{6} = \frac{SI}{c} = \frac{SLh^2}{6}, \qquad \textbf{(115)}$$

from which, the length of bearing being known, we may calculate the thickness h.

One bolt on each side is sufficient for bearings not more than 6 inches long, but for longer bearings we usually find two bolts on a side. The theoretical location for two bolts on a side, in order that the bearing may be equally strong at the bolts and at the center of the length, may be shown by the principles of mechanics to be $\frac{5}{24}$ L from each end, as indicated in Fig. 76.

The bolts are evidently in direct tension, and if equally loaded would each take their fractional share of the whole load P. This is difficult to guarantee, and it is safer to consider that $\frac{2}{3}$ P may be taken by the bolts on one side. On this basis, for total number of bolts n, equating the external force to the internal resistance of the bolts, we have :

$$\frac{2}{3}P = \frac{S\pi d_1^2}{4} \times \frac{n}{2}, \qquad \textbf{(116)}$$

from which the proper commercial diameter may be readily found.

The bracket may have the shape shown in Fig. 77. The portion at B is under direct shearing stress; and if A be the area at this point, and S_s the safe shearing stress, then, equating the external force to the internal shearing resistance,

$$P = AS_s. \qquad \textbf{(117)}$$

The same shear comes on all parts of the bracket to the left of the load, but there is an excess of shearing strength at these points.

At the point of fastening, the bolts are in shear, due to the same load, for which the equation is

$$P = \frac{\pi d^2}{4} n_1 S_s. \qquad \textbf{(118)}$$

For the upper bolts, the case is that of direct tension, assuming that the whole bracket tends to rotate about the lower edge E. To find the load T on these bolts, we should take moments about the point E, as follows:

$$PL_1 = Tl; \text{ or } T = \frac{PL_1}{l}. \qquad \textbf{(119)}$$

Then, equating the external force to the internal resistance,

$$T = \frac{PL_1}{l} = \frac{\pi d_1^2}{4} \times \frac{n_1}{2} S. \qquad \textbf{(120)}$$

The upper flange is loaded with the bolt load T, and tends to break off at the point of connection to the main body of the bracket, the external moment, therefore, being Tr. The section of the flange is rectangular; hence the equation of external and internal moments is:

$$\mathrm{T}r = \frac{\mathrm{PL}_1}{l}r = \frac{\mathrm{S}bt^2}{6}. \qquad (121)$$

It may be noted that the lower bolts act on such a small leverage about E, that they would stretch and thus permit all the load to be thrown on the upper bolts; this is the reason why they are not subject to calculation for tension.

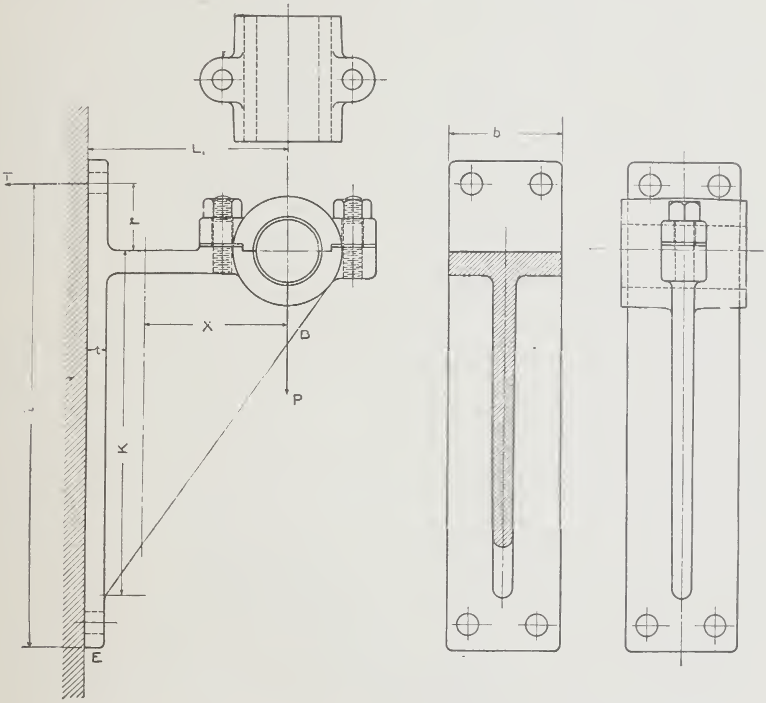


Fig. 77.

The section of the bracket to the left of the load P is dependent upon the bending moment, for, if this section is large enough to take the bending moment properly, the shear may be disregarded. It should be calculated at several points, to make sure that the fiber stress is within allowable limits. The general expression for the equation of moments is, for any section at leverage x,

$$\mathrm{P}x = \frac{\mathrm{SI}}{c}, \qquad (122)$$

from which, by the proper substitution of the moment of inertia of the section, the fiber stress can be calculated. The moment of inertia for simple ribbed sections can be found in most handbooks. The process of solution of the above equation, though simple, is apt to be tedious, and is not considered necessary to illustrate here.

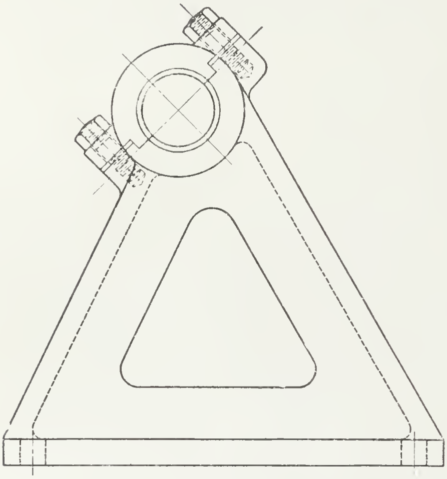


Fig. 78.

PRACTICAL MODIFICATION. Adjustment is an important practical feature of bearings. Unless the proportions are so ample that wear is inappreciable, simple and ready adjustment must be provided. The taper bushing, Fig. 79, is neat and satisfactory for machinery in which expense and refinement are permissible. This is true of some machine tools, but is not true of the general "run" of bearings. The most common form of adjustment is secured by the plain cap (which may or may not be tongued into the bracket), with liners placed in the joint when new, which may subsequently be removed or reduced so as to allow the cap to close down upon the shaft. Several forms of cap bearings are illustrated in Figs. 80, 81, and 82.

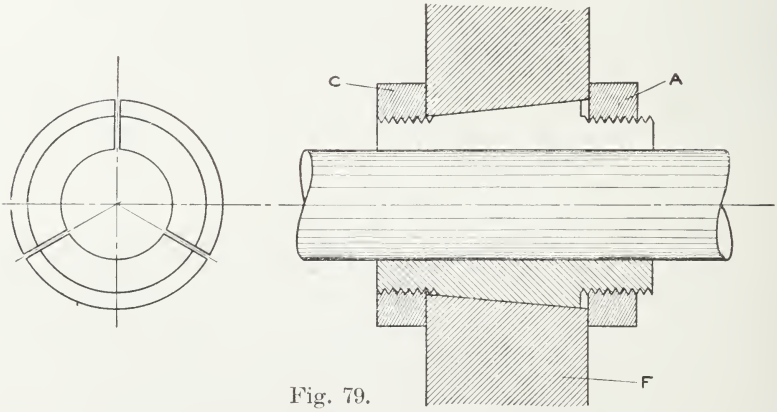


Fig. 79.

Large engine shaft bearings have special forms of adjustment by means of wedges and screws, which take up the wear in all directions, at the same time accurately preserving the alignment of the shafts; but this refinement is seldom required for shafts of ordinary machinery.

In cases where the cap bearing is not applicable, a simple bushing may be used. This may be removed when worn, and a new one inserted, the exact alignment being maintained, as the outside will be concentric with the original axis of shaft, regardless of the wear which has taken place in the bore.

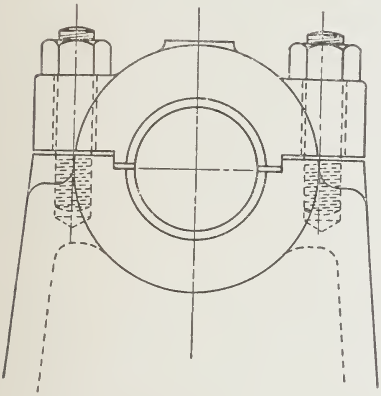


Fig. 80.

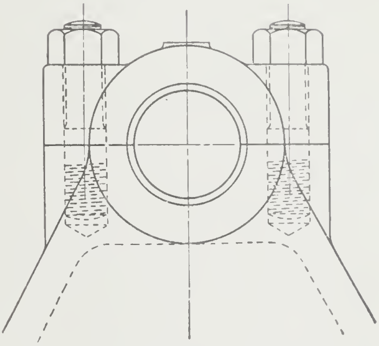


Fig. 81.

The lubrication of bearings is a part of the design, in that the lubricant should be introduced at the proper point, and pains taken to guarantee its distribution to all points of the running surface. The method of lubrication should be so certain that no excuse for its failure would be possible. Grease is a successful lubricator for heavy loads and slow speeds, oil for light loads and high speeds.

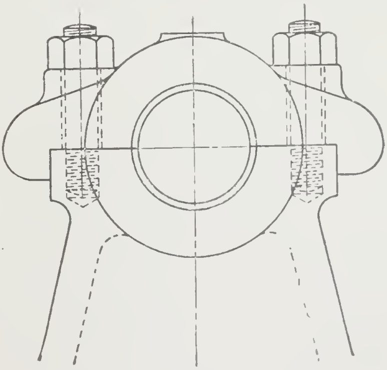


Fig. 82.

In order to insure the lubricant reaching the sliding surfaces and entering between them, it must be introduced at a point where the pressure is moderate, and where the motion of the parts will naturally lead it to all points of the bearing. Grooves and channels of ample size assist in this regard. A special form of bearing uses a ring riding on the shaft to carry the oil constantly from a small reservoir beneath the shaft up to the top, where it is distributed along the bearing and finally flows back to the reservoir and is used again.

The materials of which bearings are made vary with the service required and with the refinement of the bearing. Cast iron makes an excellent bearing for light loads and slow speeds, but it is very apt to "seize" the shaft in case the lubrication is in the least degree impaired. Bronze, in its many forms of density and hardness, is extensively used for high-grade bearings, but it also has little natural lubricating power, and requires careful attention to keep it in good condition.

Babbitt, a composition metal, of varying degrees of hardness, is the most universal and satisfactory material for ordinary bearings. It affords a cheap method of production, being poured in molten form around a mandrel, and firmly retained in its casing or shell through dovetailed pockets into which the metal flows and hardens. It requires no boring or extensive fitting. Some scraping to uniform bearing is necessary in most cases, but this is easily and cheaply done. Babbitt is a durable material, and has some natural lubricating power, so that it has less tendency to heat with scanty lubrication than any of the materials previously mentioned. Almost any grade of bearing may be produced with babbitt. In its finest form the babbitt is hammered, or pened, into the shell of the bearing, and then bored out nearly to size, a slightly tapered mandrel being subsequently drawn through, compressing the babbitt and giving a polished surface.

A combination bearing of babbitt and bronze is sometimes used. In this the bronze lies in strips from end to end of the bearing, and the babbitt fills in between the strips. The shell, being of bronze, gives the required stiffness, and the babbitt the favorable running quality.

PROBLEMS ON BEARINGS, BRACKETS, AND STANDS.

1. The allowable pressure on a bearing is 300 pounds per

square inch of projected area. What is the required length of the bearing if the total load is 4,500 pounds and the diameter is 3 inches?

2. The cross-head pin of a steam engine must be 2.5 inches in diameter to withstand the shearing strain. If the maximum pressure is 10,000 pounds, what length should be given to the pin?

3. The journals on the tender of a locomotive are $3\frac{1}{2} \times 7$ inches. The total weight of the tender and load is 60,000 pounds. If there are 8 journals, what is the pressure per square inch of projected area?

4. What horse-power is lost in friction at the circumference of a 3-inch bearing carrying a load of 6,000 pounds, if the number of revolutions per minute is 150 and the coefficient of friction is assumed to be 5 per cent?

5. The cast-iron bracket in Fig. 77 has a load P of 1,000 pounds. Determine the fiber stress in the web section at the base of the bracket if the thickness is taken at $\frac{1}{2}$ inch, and $L_1 = 12$ inches; $l = 20$ inches; $k = 11$ inches; $t = 1$ inch.

6. Calculate the diameter of the bolts at the top of the bracket.

7. Assuming r equal to 6 inches, what is the fiber stress at the root of flange?

EXAMINATION PAPER

MACHINE DESIGN.

Instructions to the Student. Place your name and full address at the head of the paper. Work out in full the examples and problems, showing each step in the work. Mark you answers plainly "Ans." Avoid crowding your work, as this leads to errors and shows bad taste. For the calculations any cheap, light paper like the sample previously sent you may be used.

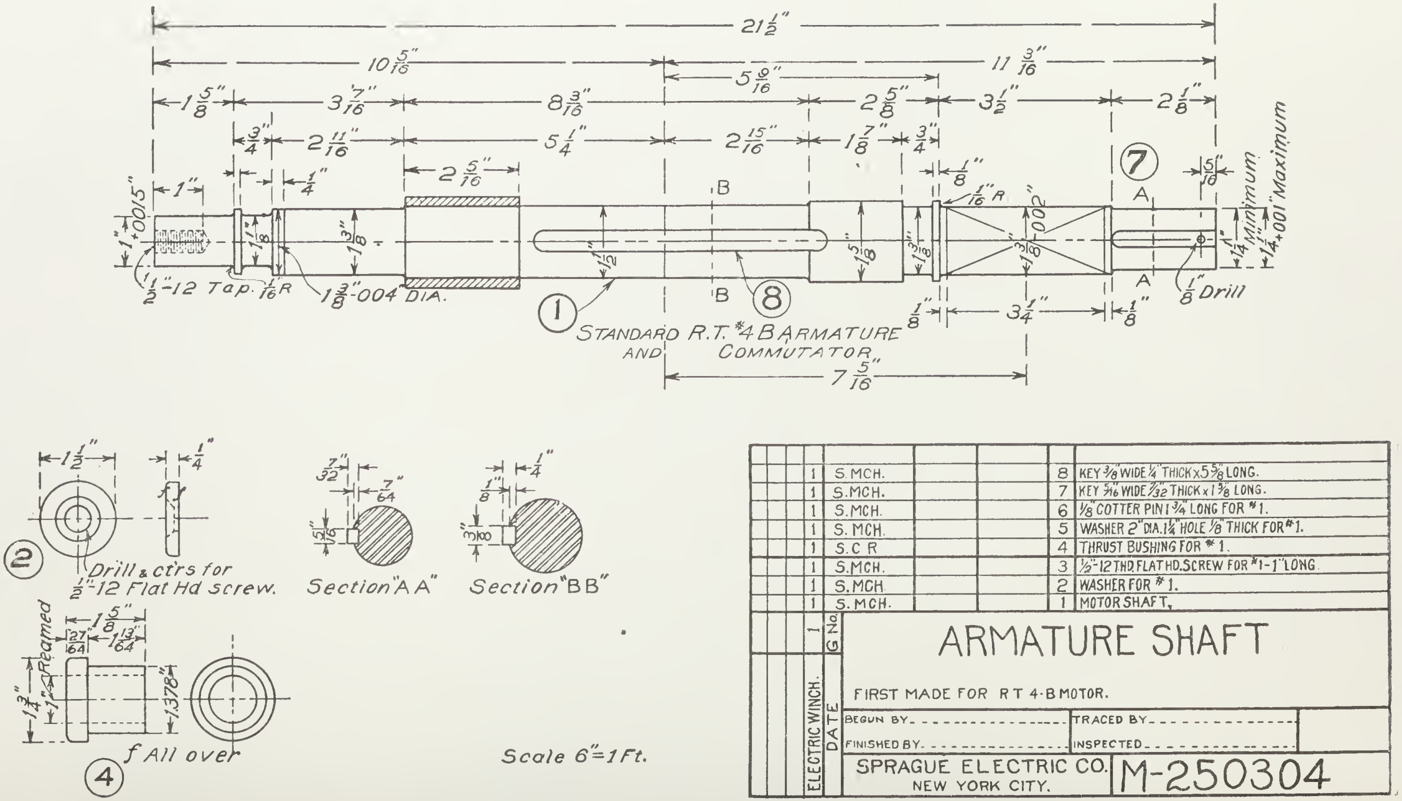
At the end of each chapter on the several subjects in Part II will be found a series of test problems intended to cover the ground of the chapter which they terminate. As a matter of *benefit to the student*, he should try to solve each full set of problems at the time he is pursuing his detailed study of the chapter. As a matter of *formal examination*, the following selected problems are required, the student being given some latitude of choice. He must confine himself strictly within the limits stated, however, and if he submits more than the required number of problems in each set, his grade will be invariably determined on the first problems read by the instructor up to that required number.

Belts	Solve	any	3	problems
Pulleys	"	"	3	"
Shafts	"	"	5	"
Spur, Bevel, and Worm Gears	"	"	4	"
Friction Clutches	"	"	2	"
Couplings	"	"	2	"
Bolts, Studs, Nuts, and Screws	"	"	3	"
Keys, Pins, and Cotters	"	"	4	"
Bearings, Brackets, and Stands	"	"	4	"

After completing the work, add and sign the following statement:

I hereby certify that the above work, including accompanying drawings, is entirely my own.

(Signed)



			1	S.MCH.			8	KEY ⅜" WIDE ¼" THICK x 5⅝" LONG.
			1	S.MCH.			7	KEY 5/16" WIDE 7/32" THICK x 1⅝ LONG.
			1	S.MCH.			6	⅛" COTTER PIN 1¾ LONG FOR #1.
			1	S. MCH.			5	WASHER 2" DIA. 1¼" HOLE ⅛" THICK FOR #1.
			1	S.C R			4	THRUST BUSHING FOR #1.
			1	S.MCH.			3	½"-12 THD. FLAT HD. SCREW FOR #1-1" LONG.
			1	S.MCH.			2	WASHER FOR #1.
			1	S. MCH.			1	MOTOR SHAFT.

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TYPICAL WORKING DRAWING OF ARMATURE-SHAFT MADE IN THE DRAFTING ROOM OF THE SPRAGUE ELECTRIC CO.

MECHANICAL DRAWING.

PART VI.

WORKING SHOP DRAWINGS.

In Parts I to IV, inclusive, the fundamental principles of Mechanical Drawing were explained and illustrated. The production of working drawings has also been discussed to some extent, and the usual characters and symbols explained and applied. The elementary work already outlined has been treated chiefly from the standpoint of correctness of line representation considered by itself, without a detailed study of the use to which the drawings so produced are to be applied.

Evidently this is the proper method, for the student should gain a thorough understanding of the principles which underlie line representation before attempting to apply them to any extended practical use. In all of this preceding work it is intended that the theoretical principles shall overshadow any incidental references made to practical application, however true and pertinent the latter may be for purposes of illustration. Hence, before taking up any advanced work, the student should fully realize the importance, in fact, the absolute necessity, of thoroughly understanding the fundamental principles which have been outlined in the books which have preceded Part VI.

At this point the student must realize that a lack of proper elementary and fundamental training will make him "go lame" at every point of his course, and probably prevent the attainment of proficiency which otherwise would naturally and almost instinctively come with advanced study. It is the thorough and ready knowledge, always at his fingers' ends, of *all* the principles of Mechanical Drawing, which makes the expert draftsman.

Plan and Scope of Advanced Work. It is now intended to throw an entirely different light on the matter, and view the subject of Mechanical Drawing from a purely practical standpoint; viz., that of Utility. It is assumed that the student understands and can use the principles which have been previously discussed.

If in a working shop drawing we choose to modify any of these theoretical principles, it will be because of increased value in *Utility* of the drawing. For example, we may desire to omit some portions of an elevation or plan or side view of a complicated casting, because certain details will thus be more clearly brought out. We may make a "zigzag" section to show construction which, by absolute fidelity to theoretical principle, would be confused, or hidden in a maze of dotted lines. We may find it convenient to place in some unoccupied corner of a drawing a layout which could not be in the least justified by any rule of projection. A multitude of transgressions like these occur on good drawings, and they are certainly justifiable from the standpoint of *Utility*, which is the true ultimate end sought for in a practical shop drawing.

These variations from the theoretical are not strictly conventionalities, because they are not classified or established, so far as we know, but are the spontaneous outgrowth, as the occasion demands, of the draftsman's purpose to make his drawing one of greatest *Utility*. He can, however, safely transgress a principle only when he thoroughly knows the principle; otherwise a blind deviation from the theoretical path will inevitably lead to difficulty.

All of the above is intended to impress the student with the idea that theoretical principles are his best, in fact, his only tools to work with; but they are not "self-hardening," like "mushet" steel; they are like the finest grade of tool steel, which must be tempered and ground and used with the best judgment of the operator, to secure the most satisfactory results.

Student Drawings. A student's early drawings are usually unsatisfactory, even to himself. Somehow they do not look like those seen in shops, and as a rule he is unable to see why this is so. Of course the difference is to some extent due to the experience of the professional draftsman. However, the superior results of the latter's work are attained largely through his systematic and workmanlike habits of execution. It should encourage the student in his early attempts to know that these essentials to the infusion of life and shop spirit into a drawing can be analyzed, outlined and grasped at the outset by earnest, intelligent effort, and really good workmanlike results obtained. To discuss, and if possible to impart these essentials of a working shop drawing to the student, is the purpose of the present book.

Essentials. The two chief essentials of a shop drawing, under which general heads a multitude of detail requirements can be summed up, are:

(1.) Absolutely complete and definite instructions from designer to workman.

(2.) Least possible cost in dollars and cents of production of the drawing measured by the draftsman's time.

It makes no difference how much we may attempt to disguise these two elements, the fact will still be apparent that "*complete instructions furnished for the least money*" is what the manufacturing shop is after, and what will be assumed as a basis for judgment as to highest commercial utility.

Completeness of Drawings. As to the first point, that of completeness and definiteness of instruction, there must be no question of degree. If the information which the drawing furnishes is positive and complete, the drawing is good. If doubt arises in the workman's mind as to what the designer intended by a certain line or dimension, or if the dimension be omitted, the drawing is bad. There is no middle ground. The instructions are either present or absent, and the drawing good or bad accordingly.

The workman of to-day is not permitted to assume dimensions or shape. It is his business to execute the draftsman's orders; it is, however, often his privilege to choose his own way of doing it, but further than this modern practice does not allow him to go. He is held as rigidly to the orders specified by the drawing as the locomotive engineer is held to his bit of tissue telegraphic order to proceed, without which he dare not enter the next block. The drawing is supreme; it is official; it must be plain, direct and all-sufficient. It is the draftsman's business to make it thus, and he is not a draftsman until he does.

This idea of positiveness must be thoroughly absorbed by the student. Positive action must be a habit which controls his every move, which marks every dimension he prints, which directs every line he draws. Every line must mean something, must have a definite reason for existence, must be necessary to illustrate the idea which he wishes to convey to the workman, and every line must be a definite measurable distance from every other line, so that its location is fixed beyond a doubt. Lines which mean nothing, and cannot be measured, have no place on the drawing; they only confuse it.

A good picture of a machine could scarcely be called to the same service as a good drawing of it. The picture might give us an excellent idea of the machine, but for the purpose of the actual construction the picture is useless, while the drawing is of positive value. This value exists simply because of, and in proportion to, the completeness of detail which it shows. Hence in making a shop drawing the picture idea is entirely subordinate to the idea of *Utility*, the latter in fact being the measure of its value.

There are certain classes of drawings—of which the Patent Office drawing is a good example—in the making of which the picture idea is predominant. Here the purpose is to illustrate mechanisms, not construct them; hence the function of the drawing is in no wise that of the working shop drawing, and as such does not fall within our discussion.

Cost of Producing Drawings. The second general element involved in producing shop drawings is their cost, as measured by the draftsman's time. It is somewhat subordinate to the first element, for the drawing must be a good one, judged by an absolute standard, whatever the time or cost necessary to produce it. Cost, however, is an important item, and cannot well be overlooked. It is inevitable that in any enterprise economy will ultimately be sought, whatever extravagance an imperative original demand may have permitted. This is as true in the production of drawings as in the case of manufactured articles of trade. Drafting-room labor is a relatively high-priced service, and the salary list easily assumes considerable proportions, so that wasteful excesses count up rapidly. One of the qualifications of proficiency invariably required for this department of shop organization is rapidity of execution. This is not as dependent upon personal traits as at first might be supposed. A man may so husband his time and direct his efforts that he will easily distance his neighbor of more rapid motion. The latter may have less ability to make his energies count, and lack of judgment as to when just enough, and no more than enough, energy has been expended on his drawings. From the standpoint of *Utility*, the function of a drawing is fulfilled when it has reached the stage that it *completely* instructs; more time spent in elaboration is wasted, and is an unnecessary and therefore extravagant expenditure. The student must fully realize this. In his earnestness to produce finished and complete work he must constantly strive to accomplish results in the least possible time. This does not mean careless haste; far from it. A complete shop drawing cannot be made by short cuts, but through a systematic building of line on line, dimension on dimension. This is in sharp contrast to a haphazard habit of developing a drawing, first a line here and then a figure there, with no definite purpose in mind, and no hint as to when the drawing is actually completed.

The one method constitutes the efficient draftsman who works easily, receives a high salary, and is worth it, because he wastes no time in unnecessary labor. The other marks his unfortunate brother, plodding laboriously far behind, receiving a small pittance per hour, and worth less, because he does uncalled-for labor, and loses his definiteness of purpose in a maze of unexplainable lines and figures.

A working shop drawing, commercially considered, may well be defined as being "*Complete instruction from designer to workman issued at minimum expense.*"

This definition should be memorized by the student, and constantly kept in mind while making a drawing. The preceding pages should be re-read with this in view until the full spirit is appreciated.

The maxim as given above, if faithfully adhered to without modification, answers nearly every question that can be raised as to the excellence of a drawing. It can be used as a standard of judgment, whatever system of lines or symbols may be in vogue. It permits a draftsman to adjust himself to the rules of any shop or drawing-room, and yet produce a good drawing and satisfy his employer.

A drawing which is cheaply produced and at the same time does perfectly that for which it was made, that is, convey complete instruction, is beyond commercial criticism.

Method of Procedure. As the general objects to be attained in a working shop drawing have now been presented, it is necessary to indicate in detail how the work may be properly accomplished. In order to do this, it is proposed to produce systematically a full set of working drawings of a familiar and comparatively simple machine. The methods used will be those of a designing detail draftsman, producing commercial work fit for shop use. In the progress of the work, from its beginning in the rough, though accurate, pencil layout, to the completion of the tracings and the order sheets, the same bold style, clearness, directness and businesslike spirit which the shop atmosphere and surroundings would naturally supply, will be emphasized, and so far as possible imparted to the student. It is expected that the student will follow the text closely and study the plates carefully, endeavoring to familiarize himself with every detail illustrated. The more closely he is able to apply himself in this respect the more will he be able to partake of the life and spirit which is intended to be conveyed, and without which the true character of the work can be but poorly developed.

Incidentally, several purposes will be fulfilled by this treatment.

Ability to read drawings quickly and intelligently is almost as important as making them, and it is expected that the study of the plates, with a view to thoroughly understanding every line, will develop proficiency in the art of reading drawings.

The discussion in the text, of not only the form of the machine parts themselves, but also the tools and shop processes to produce them, affords considerable insight into the influences affecting good machine design. Without introducing any mathematical analysis or investigation, which is beyond the province of this book, much practical consideration as to the restrictions imposed by existing shop methods upon theoretical construction will be suggested, and the student encouraged to use his judgment thereon.

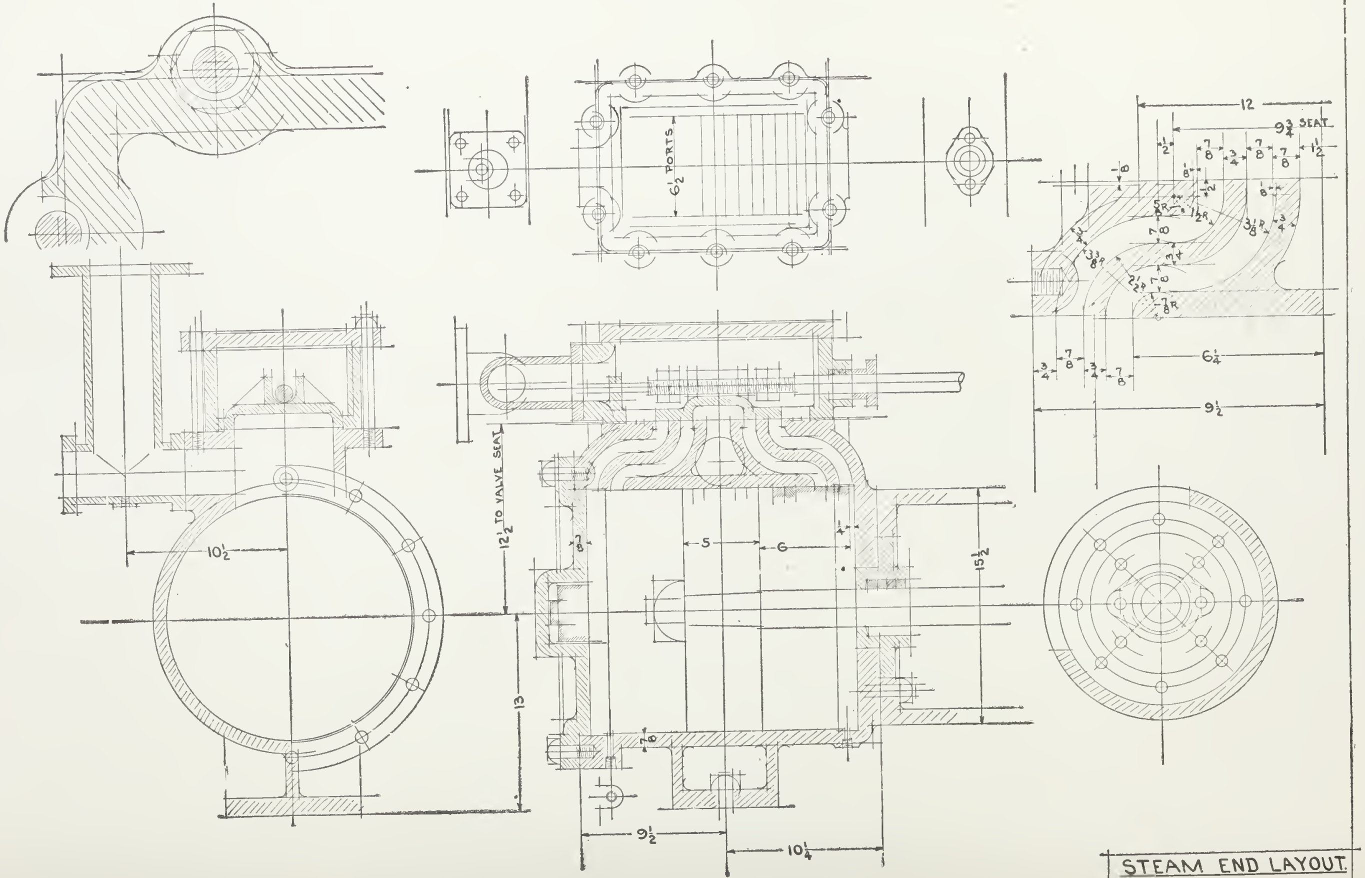
In the preliminary layouts the actual "sketchy" appearance of the pencil drawing will be imitated as far as possible, so that the student himself may imitate and catch the bold dash, yet fine accuracy, of the line-work, which is characteristic of the expert draftsman.

The completeness of a *set* of drawings is as important a lesson as the completeness of each drawing itself. In this is involved the proper arrangement and classification of details, the foundation layout, and the system of order sheets for getting work into and through the shops. This is a feature which very strongly affects some of the finishing touches to a drawing, for it is so easy to omit a "few last things" and turn in an uncompleted sheet. Every draftsman knows how many little things come up toward the close of a job involving complete drawings of a machine, and how strong the tendency is to omit them, and relieve himself of somewhat tedious details. The result is irritation and delay when the drawings get into the shop, and they return to the drawing room to be fixed up at a time probably inconvenient for all parties concerned. A good draftsman will turn in a *complete* set of *complete* drawings. It is highly important that the student grasp this idea, and study his work accordingly.

DUPLEX PUMP PLATES.

The typical set of plates chosen for this book in fulfillment of the above purposes, takes up the study of a simple, duplex steam pump. This particular type of machine represents the simplest and most elementary form of the steam engine in modern use in respect to valve gear and controlling devices. It is not an economical machine, yet its principles lie at the foundation of the economical high-speed engine, the latter being produced through a modification of the uneconomical valve gear such as is found on a pump of the type chosen, rather than through any radical change of construction as to the body of the machine. Hence the study of a steam pump may well precede that of higher forms of the steam engine. It is hoped that the study will so interest the student that he will be led to further investigation and development not only of the steam engine itself, but of that highly important division of modern engineering,—pumping machinery.

Thus we note another point of advantage in the study as out-



STEAM END LAYOUT.
16-8½-12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
BOSTON-MASS.
SCALE-3=1'

Ⓐ

lined. The power end of the machine introduces us to the steam engine; the load end is the beginning of the engineering of pumping machinery.

Rating of Pump. A steam pump is rated by the bore of its cylinders and length of stroke, all being given in inches. A "$16 \times 8\frac{1}{2} \times 12$ pump" means that the steam cylinder is 16 inches in diameter, the water plunger $8\frac{1}{2}$ inches in diameter, and the nominal length of stroke 12 inches. These sizes are always given in the same order, beginning with the diameter of the smallest cylinder (in case there is more than one), then the diameter of water plunger, the common stroke of both being placed last. This expresses to the mechanic the rating of the pump in the clearest style and briefest language.

The pump illustrated here is designed for standard service, operating under a steam pressure not to exceed 100 pounds per square inch, water pressure not to exceed 150 pounds per square inch, and the rated capacity based on an average piston speed of 100 feet per minute being about 550 gallons. This requires that each side of the pump shall handle 275 gallons, and being double acting, shall make 100 reversals or 50 double strokes per minute.

Plate A. Steam End Layout. This plate illustrates, as nearly as reproduction can accomplish, the pencil layout of the steam end. It is the first work of the designing draftsman. The drawing as shown is exactly the type of layout which he would turn over to a detail draftsman, whose duty it would be to work up detail shop drawings therefrom.

The character of this drawing should be carefully studied. Remember that it is a layout, nothing more; also bear in mind that it is an exact, measurable working sketch. Attention is called to the sharpness of the lines, especially to the clean-cut intersections. Note the boldness, dash and businesslike style, the freehand cross-section lines roughly put in. There is no hesitation or worry as to where the end of a line shall be, or whether it crosses other lines which it theoretically should not. The intersections are allowed to indicate the termination of lines, and the rough section lines pick out the parts and separate them clearly to the eye. There is the spirit in this layout of confident, definite and rapid action, with no thought for absolute finish in line-work, but with every thought for absolute results as to measurable dimensions.

The data for the production of Plate A by the student are rather more complete than he would usually find in practice. Plates E, F and G show many details fully.

The steam cylinder and head, however, as shown in Plate G, are not dimensioned, and the student's problem is to produce this plate complete, with finish marks, dimensions and necessary data for a working drawing. In order to do this it is first necessary to work up Plate A with exactness, in pencil, and see that all parts go together properly. Then the detail of cylinder and head may be made separately by measurement of the layout drawing, and Plate G produced.

For this work the ordinary brown detail paper is very satisfactory. A hard lead pencil is necessary, as hard as 6H, and the point must be kept well sharpened.

There are two general rules of action in producing a drawing which give the answer to the question which oftenest confronts the beginner: "What is to be done first?" or "What is to be done next?" These rules are:

1. Draw everything that is positively known.
2. Work from the inside to the outside.

Every problem has some positive data, assumed or calculated, to start with. The first thing to do in every case is to get this data represented by lines on the paper. An expert designer has been heard to say that until he had spoiled the blankness of his sheet of paper by some lines, he could not design. There is something in this; and almost invariably the first line to draw is a horizontal center line somewhere near the middle of the sheet; draw it! Draw it at once without hesitation, and the layout is begun. We now have something about which to build.

In this case the designer would first calculate the size of the piston rod, and determine the fastening to the piston. He would then draw the rod and build a hub around it. He would next calculate the width or thickness of piston and size of packing rings, and draw the two vertical lines 5 inches apart, to indicate the piston faces. These lines would be limited by the cylinder bore, which he knows to be 16 inches; hence horizontal lines 16 inches apart, parallel to and symmetrical with the center line, are the next to be drawn. Short vertical lines indicate the location of the packing rings. As the nominal travel of the piston is to be 12 inches, the location of the piston and rings can be shown on both sides of the central vertical line at the limits of travel. A clearance must exist between the heads and the piston (in this case $\frac{1}{4}$ inch is allowed), hence the lines of the heads can be drawn, and the general inside outline of the cylinder barrel is complete.

This is all in direct application of the foregoing rules, and is so simple, natural and direct that it hardly requires such explicit statement. We have simply taken such data as we had and put it on paper, placing it where it can be seen from all sides, and where the mind is relieved of the labor of carrying it.

If the student will only appreciate this one rule and draw all he knows about the problem, he is well on his way to its solution. *Draw everything you know, and work for what you don't know*, is what these two rules say, and the first question to arise should be: "Have I drawn everything that is known about the problem?" before he asks himself or any one else: "What shall I do next?"

One other rule might be added to these two: *Keep dimensions in even figures, if possible.* This means that small fractions should be avoided. It is just as easy to bear this point in mind, and save the workman much annoyance and chance of error, as it is to disregard this matter. Even figures constitute one of the trade-marks of an expert draftsman. Of course a few small fractions, and sometimes decimals, will be necessary. Remember, however, that fractions must in every case be according to the common scale; that is, in sixteenths, thirty-seconds, sixty-fourths, etc.; never in thirds, fifths, sevenths, or such as do not occur on the common machinist's scale.

A systematic, definite mode of treatment on these lines must become a habit, so that all problems, however complicated, can be approached with confidence in the same way. It is the drawing of one line which makes clear the drawing of the next and subsequent lines; and the most serious obstacle which the student is likely to set for himself is trying to see the whole problem through from the beginning. Even an expert cannot do this, but allows the layout to develop results as he proceeds.

The details of the piston and rod being given in Plate E, the foregoing work is very easy for the student. The thickness of the barrel and heads being determined ($\frac{7}{8}$ inch in this case), the exterior outline may be partially drawn. The fixed head at the yoke end must be thicker than this, in order to receive the yoke and stuffing-box bolts without breaking through. The recesses or counterbores at either end of the cylinder should be so located that the packing rings run over the edge a little at the end of the stroke, thus preventing the wearing of a shoulder by the piston stopping in the same place every time. The counterbore should be deep enough to allow reboring the cylinder without the counterbore being touched by the tool. In this way the counterbore is retained to center the cylinder at its original location.

The size of steam ports having been calculated, they may be drawn in, the turns being made easy and as direct as possible. The height to valve seat must be kept at the lowest limit consistent with sufficient metal between and outside of the ports. As the detail of the ports might be somewhat troublesome, it is shown in an enlarged sketch for the student's benefit. Chipping or filing strips $\frac{1}{8}$ inch high are left on the port edges, which must be true, in order to finish them up easily.

The three inner ports are for exhaust, the outer ones for admission of steam. This five-ported cylinder is peculiar to the direct acting steam pump, it being a device to effect the cushioning of the piston at the end of the stroke, thus preventing the piston from striking the heads. This is necessary, since no positive limit of motion exists, as is the case in machines with crank and connecting rod.

When the edge of the piston has passed the outer edge of the exhaust port, as shown in Fig. 1, the steam, which has been exhausting through port A, is confined in space B and port C, and, being compressed by the piston, acts like a spring to retard its motion. If the point P is properly determined for a given speed, the piston will always compress the steam just enough to cause it to stop at the end of the nominal stroke; in this case, $\frac{1}{4}$ inch from the head. It is evident, however, that at different speeds the piston will have more or less power to compress the steam, and will not stop at the point desired. This causes the trouble of "short stroke," and consequent inability to make the pump work to its full capacity. Now if we connect ports A and C by a small opening shown dotted at D, and control this opening by a plug valve operated by hand from the outside, we can let a little steam leak by into port A, thus reducing the cushion and allowing full stroke.

In order to avoid complicating the drawing, no cushion valves are shown or required to be put on by the student. They are not customary in small pumps, but might advantageously be put on the present illustration.

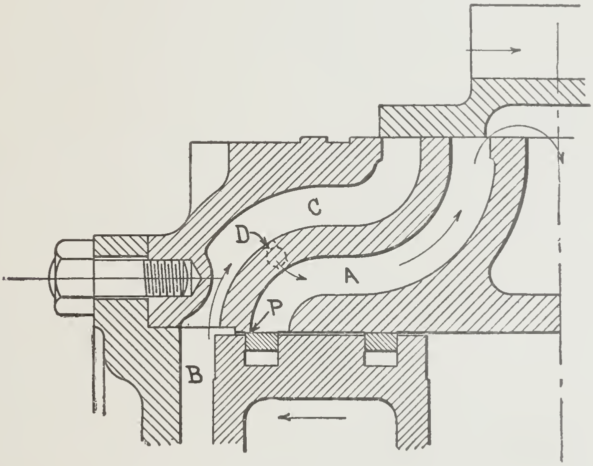


Fig. 1.

The valve seat must be a scraped surface, while the chest face need not be; hence the latter is finished $\frac{1}{8}$ inch lower. This also gives a ledge against which the steam chest fits, thus securing positive location.

The bolting of the heads and the steam chest should allow a width of packing inside of the bolts of $\frac{1}{2}$ to $\frac{5}{8}$ inch, otherwise there is danger of the steam blowing out the packing and causing leakage around the bolts. The bolts do not fill the holes, the latter being drilled large, from $\frac{1}{16}$ to $\frac{1}{8}$ inch. The spacing, if wider than 5 or 6 inches, is likely to permit springing of the flanges between the bolts, and consequent leakage. Bolts less than $\frac{5}{8}$ inch diameter are not desirable, as they can be easily twisted off with an ordinary wrench. In this case the cylinder head takes $\frac{7}{8}$-inch bolts, the yoke, stuffing-box and gland, $\frac{3}{4}$-inch.

The flanges of heads and cylinders are usually from 25 per cent to 50 per cent thicker than the body of the casting.

Drips, $\frac{1}{2}$-inch pipe tap, to be fitted with cocks, are necessary at both ends of the cylinder to readily drain the cylinder of water.

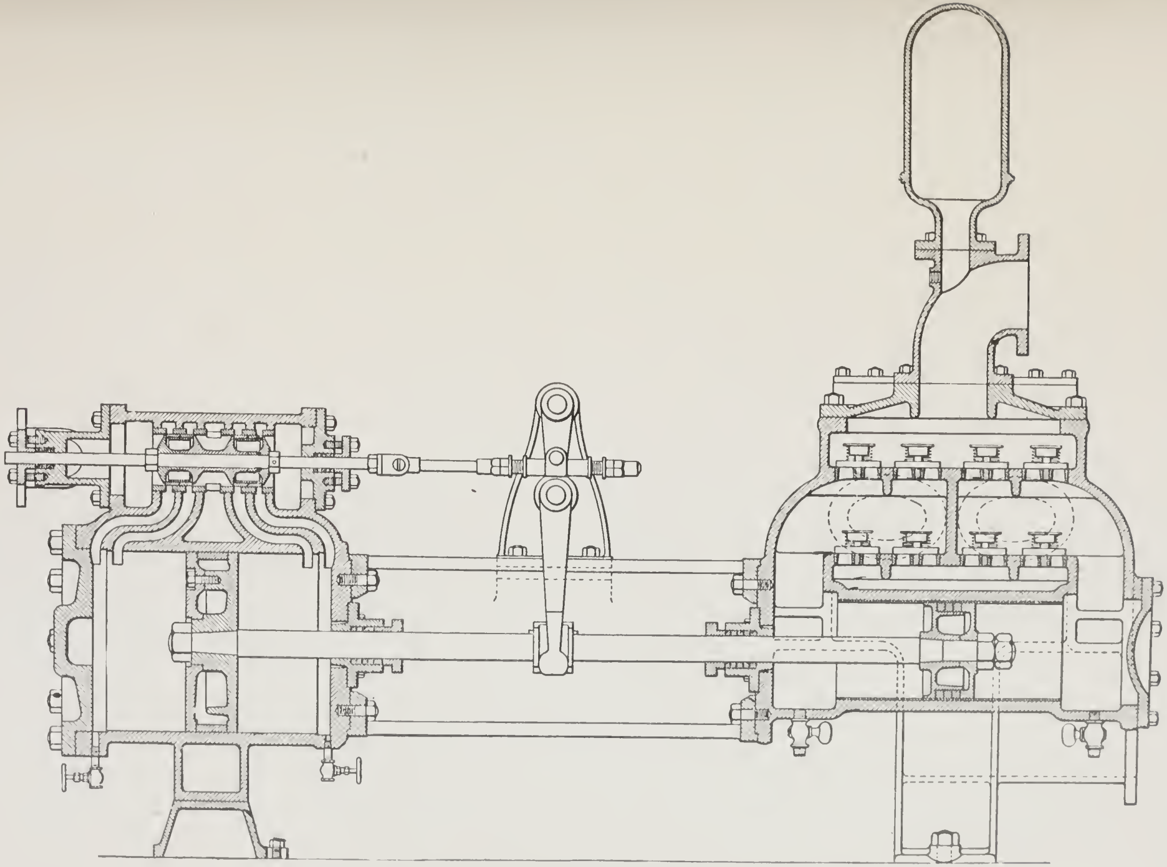
Molding of Steam Cylinder. The design is often influenced by the way in which the piece is to be cast. It often takes but a slight change of design to save many dollars in pattern making and foundry work. Hence the habit should be formed of always judging the design of a piece from the foundry standpoint. In this case it is evident that the ports and cylinder bore must be cored out, and the most obvious position of molding is to lay the cylinder on its side, the parting line of the flask being along a vertical plane running lengthwise through the middle of the cylinder. This permits the chest flanges to draw nicely, likewise the ribs on the foot, and allows the thin curving port cores to stand edgewise in the mold.

Another method of molding would be with the valve seat down. This would involve loose pieces for the chest flanges, and setting of cores for the cylinder foot. It would, however, assure sound metal beyond question at the valve seat. Spongy metal at the important wearing surfaces, the valve seat and cylinder bore, is not permissible in any case, and care in molding, and good design, is necessary for good results.

All corners must be carefully filleted, and chunks of metal must be avoided, especially where several walls or ribs join. The metal must be kept of average uniform thickness, so that the whole casting will cool uniformly.

Machining of Steam Cylinder. The boring may be done on a vertical boring mill, the heavy arm carrying the tool being thrust down unsupported into the cylinder, the latter being rotated by the table to which it is clamped. If the horizontal boring machine be used, the hole through the inside head for the stuffing box must be large enough to permit a stiff boring bar to be passed through. This allows a support at each end of the bar, to take the strain of the cut.

The plane surfaces may be finished on a reciprocating planer



CROSS SECTION OF DOUBLE ACTING STEAM PUMP.

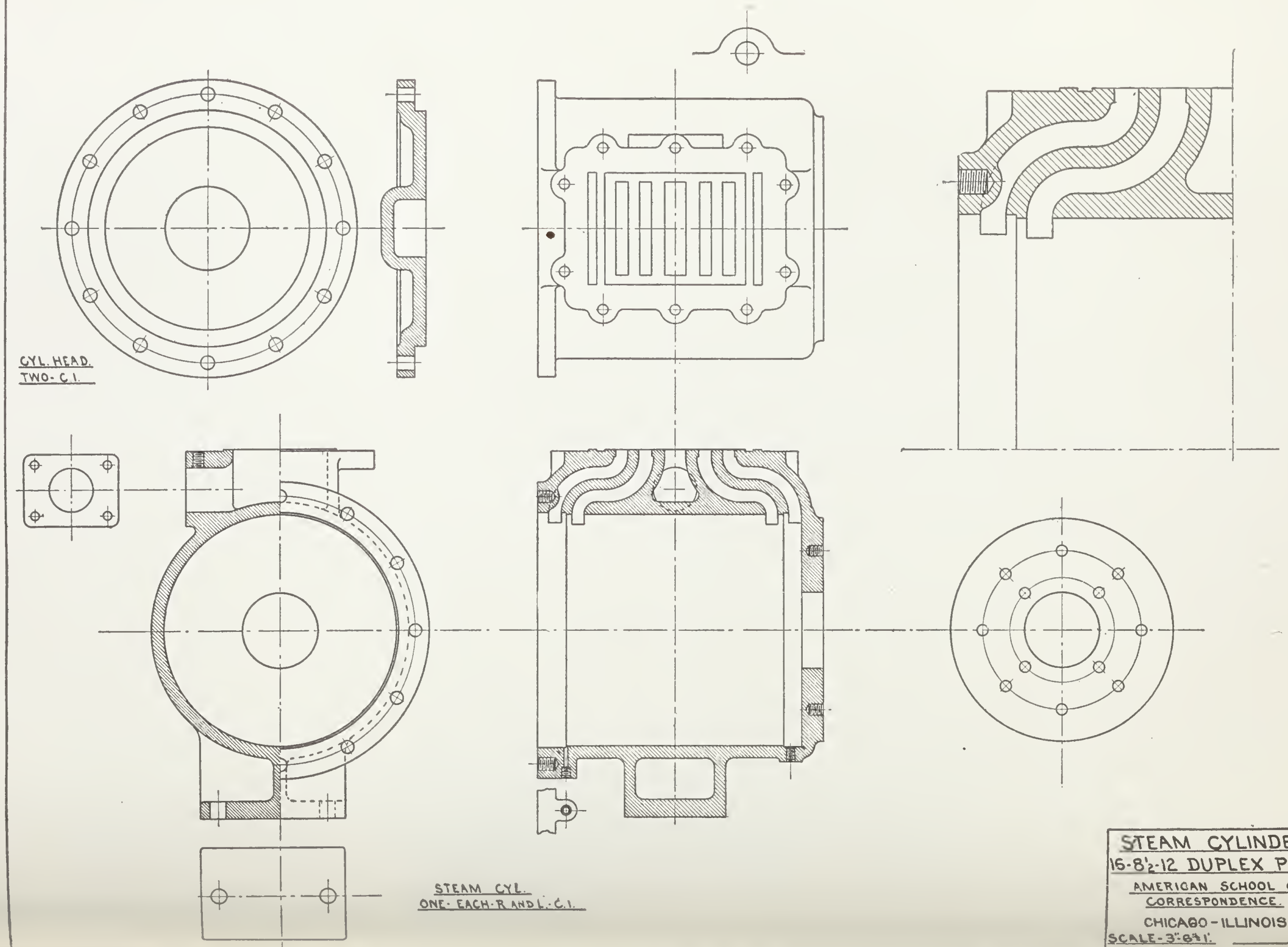
or a rotary planer. In the latter case it is desirable to keep all lugs or projections back from finished surfaces, to permit the large round head which carries the cutters to pass over them without interference.

The drilling of standard machine parts of this character is usually done through jigs, or plates carrying hardened steel bushings laid out to correspond with the holes required, and through which the drill is guided. These plates are located by some fixed line or lug on the casting, and then clamped fast, thus assuring exact duplication and rapid drilling, and avoiding the tedious laying out of the holes. In order to save changing the drill it is desirable, if possible, to maintain the same size of hole on any given surface. Of course it is not always admissible to do this.

Plate G. Steam Cylinder. After the exact and complete development of the steam-end layout, the student should be pretty thoroughly acquainted with the details of the cylinder. All the work thus far has been entirely for *his own* information, to get his ideas in visible shape, so that he himself can have a permanent record of them. This layout, however, is not in suitable form to finish up into a detail drawing. Its sketchy nature and the confusion of parts, especially if attempt were made to add dimensions, would render it somewhat difficult to be read by a workman taking it up as an unfamiliar subject. Hence it is now necessary to separately detail the parts, with the object in view of transferring, in the simplest and most direct manner, specific information to the workman which will enable him to construct the several parts. It is not enough now that the drawing be clear to the man who makes it; it must be clear, *absolutely clear*, to the shop mechanic, who has no means of knowing the designer's plans except through the information which the drawing gives on its face.

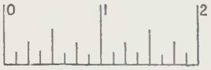
This requires that the draftsman should put himself in the workman's place, and forestall, by the explicit nature of his drawing, all possible questions which may arise in the shop. In this way only can he hope to avoid errors of construction and the continual annoyance of endless explanation of his orders.

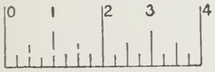
Plate G is to be a finished drawing, and the first thing to do is to lay out the sheet. The standard sheet for details which has

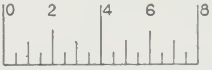


STEAM CYLINDER.
16-8½-12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE - 3" = 1'.

been adopted is 18 × 24 inches trimming size, with $\frac{1}{2}$ inch margin all round, so that the working space is 17 × 23 inches. The rectangle for the title is to be laid off $2\frac{1}{2} \times 4$ inches in the lower right-hand corner, and must never be altered, either in size or position. This does not mean that other sizes are wrong, but once a standard system is adopted it must be strictly adhered to, both for artistic and commercial reasons. The scale to which the drawing is to be made is indicated in the title corner on every plate.

The scales permissible for shop drawings in the United States are those readily derived from the common foot rule, such as full size, 6 inches = 1 foot, 3 inches = 1 foot, $1\frac{1}{2}$ inches = 1 foot. These are the most common, most easily read from an ordinary scale, and one of these can usually be adopted. The student should learn to read these from an ordinary scale without being confined to a special graduation. To do this it is not necessary to divide each dimension by 2, 4 and 8 to get half size, quarter size, or eighth size, and then lay down the result. For half size, or 6 inches = 1 foot, $\frac{1}{2}$ inch on an ordinary rule represents 1 inch. Hence, each half inch may be read as 1 inch, and its subdivisions accordingly, thus: 

For 3 inches = 1 foot, or quarter size, $\frac{1}{4}$ inch represents 1 inch, and looks thus: 

For $1\frac{1}{2}$ inches = 1 foot, or eighth size, $\frac{1}{8}$ inch represents 1 foot, and looks thus : 

It is very easy to get accustomed to this, and it saves much time and trouble hunting up a special scale every time.

The other allowable scales, less common, but sometimes necessary on large work, are 1 inch = 1 foot, $\frac{3}{4}$ inch = 1 foot, $\frac{1}{2}$ inch = 1 foot, $\frac{3}{8}$ inch = 1 foot, $\frac{1}{4}$ inch = 1 foot, and $\frac{1}{8}$ inch = 1 foot. To use these scales conveniently, special graduation is desirable.

The general arrangement of the sheet, number of views and approximate space occupied, should be blocked out first. This can easily be done from the original layout. In general, several cross-sections are preferable to a single view, which involves many dotted lines. Dotted lines are very convenient for showing invisible parts of an object, but they are often abused, and the drawing of a complicated piece made indefinite and confused thereby. As already stated, a working shop drawing is solely to convey information to the workman at the least possible cost. A careful consideration of this will settle the question of the number of views necessary, their character, and the amount of dotted line work desirable.

Never let the drawing become the master; always be master of the drawing. Do not draw an extra view if no use can be seen for it. Do not put in dotted lines if the detail is completely shown without them. Full lines, or lines which show visible portions must, of course, be shown completely.

The nature of the pencil work on Plate G should be the same as on the original layout; viz., sharp, definite lines and positive intersections. Above all things learn the habit of accurate workmanship, for it will save many errors and a vast amount of time. The draftsman must check himself at every line he draws. Slight errors in scaling will often throw parts out of proper relation to each other, and interferences, which the drawing does not show, will become apparent only when the parts get into the machinist's hands.

It is dangerous practice to project across from one view to the other. It only takes a slight irregularity or spring in the T-square to vary the location of lines very perceptibly from where they should be, and once out of scale from this reason it is almost impossible to work a view with any certainty. Rather than project across from view to view, the principal lines, at least, should be scaled off on each view, and it will be found that in the end time will be saved and greater accuracy secured.

It is not economical of time to finish one view before beginning another. It is better to take some single detail of the drawing and develop it in all views, in order to study it from all sides. What is completed in one view may be found to be totally wrong when developed from another side, and the time spent on the first view will be wholly wasted. For example, in the present case the steam ports should be drawn in side elevation, end elevation and plan, and when thus completed the mind can leave them and in a similar fashion take up the study of the flanges, then the cylinder foot, and so on. Thus again the draftsman is master of his drawing, for he is continually making it tell him whether he is right or wrong. If, on the contrary, he allows himself to look at but one side at a time, and works from that standpoint alone, it may lead him into many difficulties from which he cannot readily extricate himself.

Do not be afraid to use the eraser. The draftsman who hesitates to draw until he is positive that no change will be necessary, is likely to spend the greater portion of his time in unprofitable dreams, for he is attempting the impossible. A drawing is a means, not an end; and, as has been already pointed out, it greatly assists the draftsman in clearing up many doubtful questions which the imagination alone cannot do.

A bold attack of a problem shows the quickest path to its solution, even if lines must be erased again and again. It is a sign of serious lack of ability to hesitate in the use of pencil and eraser.

Attention is called to the simple, straightforward character of Plate G. Notice the almost entire absence of dotted lines; the enlarged section through the ports, giving ample opportunity for dimensions without confusion; the use of a half end elevation and a half cross-section,—the one to make clear the flange and bolt layout; the other to show the exhaust opening, the small auxiliary views (drawn at convenient points) of the exhaust flange layout, the cylinder foot and the drip boss.

A steam cylinder is a fairly complicated casting; and it would be an easy matter, by the use of elaborate views, the dotting in of parts already completely shown, and careless line work, to rob this drawing almost entirely of its clearness and directness of illustration. *Just what is necessary* (for clearness' sake) *and no more* (for cheapness' sake), is the whole matter in a nutshell, and is what determines its shop and commercial value.

Dimensions and Letters. A good line drawing can be spoiled by poorly arranged dimensions and hasty lettering. The five principal points to be kept in mind to develop excellence in this respect are:

(1) System.
(2) Accuracy.
(3) Clearness.
(4) Completeness.
(5) Character.

System. The habit of system in placing figures and letters on a drawing is the one element which, to a large extent, controls all the others. If the systematic habit is established early, the other requirements will be fulfilled more easily. A haphazard method will, on the contrary, just as surely prevent the successful cultivation of the ability to figure a drawing. In fact, if the haphazard habit is continued it will itself, by the dissatisfaction which it causes, soon compel the draftsman to change his occupation.

In the first place, whatever part of a machine detail is to be dimensioned, that particular part should receive attention until it has been completely figured. Do not jump from one point to another, putting in a figure here and another there. Stick to one thing until it is done.

For example, take Plate F and the simple detail of the steam pipe. Suppose we start with one of the square flanges. The first question is: "Where is this flange located?" This is answered by the dimensions 5 inches and 21-inch centers, which refer the face of the flange to the center of the pipe and the flanges to each other. The next question is: "What are the three dimensions of the flange,—length, breadth and thickness?" This is readily answered as shown on the drawing. The next question is: "What further description is necessary to completely specify the shape of the flange?" This is answered by the radius of the corners, $\frac{3}{4}$ inch R. Next, "What drilling or special feature exists in the flange?" This is answered by $\frac{11}{16}$-inch drill, $3\frac{1}{2}$-inch centers, and the letter *f* to denote that the face is to be finished.

The round flange of this pipe is approached and figured in the same way, except that the location of the face is preferably referred to the face of the square flange by the figure $8\frac{1}{4}$ inches, instead of to the center of the pipe, because the planer hand will more naturally use this figure.

These flanges are now to be connected by a pipe involving two sizes. The main pipe is 3 inches diameter inside, 4 inches outside, and $\frac{1}{2}$ inch thick, running into the two branches by fillets and radii, as figured. The two branches are really one pipe, $2\frac{1}{2}$ inches inside, $3\frac{1}{2}$ inches outside, $\frac{1}{2}$ inch thick, and sweeping down into the square flanges by 4-inch radius.

This systematic method takes longer to explain than to actually execute, but it is typical of the train of thought which must be followed on all pieces, simple or complicated, in order to properly place dimensions.

In general, it may be stated that all parts of a piece must be referred either to each other, or to some common reference line, or to both. Each part so referred must then be figured as a piece by itself, and then its connections to the principal structure. Thus, figuring a machine detail involves three things:

(1) Relative location of its parts.
(2) Proportions of these parts.
(3) Proportions of connecting members.

As in the original design of a piece so in the figuring of it, the draftsman must as far as possible put himself in the place of the workman, judging the methods and processes of construction and available tools. This will largely influence the arrangement of the dimensions. Of course it implies considerable experience in shop work, which some students do not possess. He can begin none too early, however, to learn to look at his work from the shop standpoint, and surely make it some better on that account.

Pieces must not only be systematically dimensioned, but regularly specified and called for by suitable titles.

A title should specify at least three things:

(1) Name of piece.
(2) Number wanted for one machine.
(3) Material.

To these might be added a fourth; viz., pattern or piece number. The latter is not specified on the drawings under discussion, because systems of pattern and piece numbering are so varied that little would be gained by developing one for this special study.

These titles should always be put on in the same way, as the workmen become used to a certain system and are likely to misunderstand directions if a regular plan is not followed. A good way to arrange titles is suggested on the plates, although there are others which might be used.

Bolts are usually specified by diameter and length under the head, the length of thread being to some standard system in use by the shop, unless otherwise called for. Bolts are specified on the sheet containing the piece into which they are tapped. In the case of through bolts, tapped into neither piece, they are preferably called for in connection with the principal member.

Accuracy. Of course the dimensions on a drawing must be accurate. It is, however, a very easy matter to make errors. To insure accuracy a figure must *never* be put down carelessly, and a constant watch must be kept that scaled figures add up to over-all dimensions. It will not do to rely on scaling alone, as a very slight variation from exact scale may throw two dimensions out with each other. In spite of all the care that can be exercised errors will creep in, and a final thorough checking must be given a drawing before it is pronounced complete. A good rule to follow in checking up is to "assume everything wrong until it is proved to be right."

Clearness. As in the line drawing itself, there must be absolute clearness of instruction by the dimensions. Any doubt as to what a figure is, or what it means, rules out that figure as part of the drawing. If a piece is made wrong because doubt of this character is transmitted to the workman, the draftsman is always held responsible for the error.

Figures should, in all cases, be placed where they can be most clearly read. They should be bunched on a single view as far as possible, but not when greater clearness demands that another view be used. It hinders the reading of a drawing materially if the eye is forced to jump over large spaces of the sheet from view to view, to catch the several dimensions of a small detail. Usually it is easy to so group figures as to avoid this.

It is a good plan to keep dimensions off the body of the drawing, when it can be done so conveniently. It is not worth while, however, to go out of one's way to do this, as figures in the open spaces of a detail do not at all destroy its clearness.

Extended notes on a drawing to make it clear should not be required, but they should be used without hesitation if any doubt exists. An explicit note of instruction is the final resource for clearness when the art of drawing fails of its purpose, as it sometimes does.

Completeness. A detail is completely dimensioned when it shows *all* the figures necessary for the workman. Anything short of this is incompleteness. As modern shops hold the draftsman solely responsible for the design, the mechanic is not allowed to modify it by filling in any omitted dimensions. The only way to be sure that all the dimensions are on is to systematically go all round a piece inside and out, according to the method suggested under the paragraph on "System."

It is a good plan to always bear in mind that not only the machinist is to use the drawing, but also the pattern maker. For the benefit of the latter, special attention is desirable in figuring the cores. This saves him some addition and subtraction. In general, it has been found that less chance of error exists if mathematical work is not required of the shopman, all necessary data being furnished on the face of the drawing.

Character. By character in figures and letters is meant uniform style, height and slope, and a certain boldness peculiar to the work of the expert draftsman. The last is difficult for the novice to acquire. The student should not be discouraged because his efforts do not look like impressions from printers' type. Artistic excellence is the result of long experience, but is based on character. If the student can once get character into his work, the artistic feature will, with careful and constant practice, gradually develop. It is safe to say that there is no one element of a drawing which more positively stamps it as the work of an amateur than the character of the lettering, and every attention should be paid to getting out of the apprenticeship stage in this respect. Freehand lettering only is permitted in the drawings illustrated herewith. Ruled letters are seldom found on any working drawings, as the element of time involved is so great that few shops are willing to pay for it.

Uniform style requires that if capitals only are used in titles, they only must be used in notes and elsewhere on the drawing. If lower-case letters are used, they must be used in every part of the drawing. One style should not be mixed with another. The height of the letters should be limited by two horizontal lines, and though practice may render the upper line unnecessary, it takes but an instant to draw it, and uniform height is then assured. A good height for titles of details such as are illustrated is $\frac{5}{32}$ inch. The height once chosen should be adhered to throughout the whole set. A medium, not a hard, grade of pencil (3H) will give the hand greater freedom. A great temptation exists to omit titles from the pencil drawing, simply inking them on the tracing. This is false economy of time, for in the end it will be found that enough time will be saved by the certainty with which the tracing can be made to more than pay for the labor on the pencil drawing. Again, it permits the tracing, in regular shop practice, to be made by cheaper labor than that which produced the pencil drawing.

Uniform slope is most easily acquired by the use of guide lines put in at frequent intervals. A small wooden triangle can be made, giving the required angle. The angle of the letters shown on the plates is 9 degrees, or about 1 inch slope in 6 inches. The question as to whether letters should incline backwards, forwards, or stand vertical, does not enter this discussion. Character is not affected by the slope. The student may choose whatever comes most natural to him, but having chosen, the character of his work will be spoiled if he varies it. The most difficult of the three is the vertical style; hence most draftsmen incline their letters. The backward slope is used on the plates of this shop drawing paper, thus giving the student opportunity to compare with plates in the earlier books, and follow his preference.

The effect of change of style, height and slope is shown in Figs. 2, 3 and 4, respectively. Attention is called to Fig. 5, which is a sample title, in which these points are corrected.

Principal Titles. The principal title of a drawing should contain at least seven items:

(1) Name of principal details shown.
(2) Name of machine.
(3) Firm name and location.
(4) Scale of drawing.
(5) Date of completion.
(6) Draftsman's signature.
(7) Filing number.

To these are often added others, but for purposes of filing and reference the above at least *must* be put on. The filing number may or may not be put in the title frame, but it is really a part of it. It is often put in the margin below the title.

An arrangement of title should be established and then followed exactly, without variation either as to location on sheet or detail make-up. Abbreviated words are always permissible in titles, provided the meaning is clear. Special care must be taken in punctuation, however, as a title, whether abbreviated or not, has an unfinished appearance if the periods, commas and other necessary punctuation marks are not included.

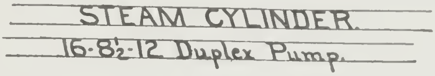


Fig. 2.

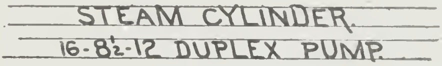


Fig. 3.

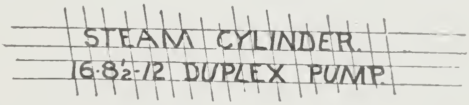


Fig. 4.

The sample title illustrated in Fig. 5 indicates the arrangement chosen for the drawings of Part IV. Note that it is necessary in this special case to add an extra subject to the seven given above; viz., the residence of the student draftsman.

This style of title must be put with care on every drawing, even on the rough pencil layouts. In the latter case it may of course be left in pencil, as the rough layouts are not to be inked.

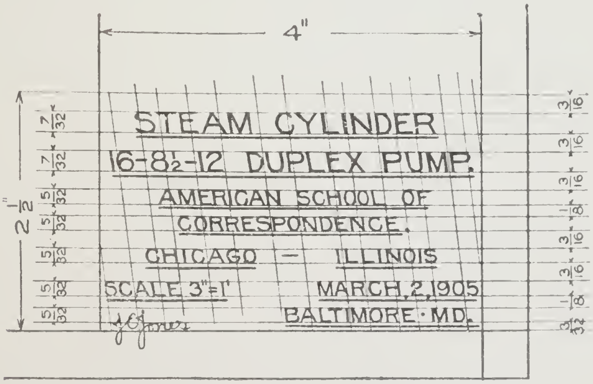


Fig. 5.

Inking and Tracing. Both bond paper and tracing cloth are used in business practice for finished drawings. It is desirable to keep a stock of both in any drawing office, so that either may be used as occasion requires. Bond paper stretched on the board gives a beautiful surface to take the ink, and very handsome and effective detail or assembled drawings can thus be produced.

Changes are not quite as readily made on bond paper as on tracing cloth, and it takes a little longer to make the blue print. In other ways the bond paper is not quite as flexible to use as the tracing cloth. However, one must be guided entirely by shop conditions to settle the question of preference. As the tracing cloth is generally used, and suits the purpose of the student better, it will be required in this work.

The inking should be done on the *rough* side of the cloth. One reason for choosing this side is that as the cloth tends to curl under toward the glazed side, the drawing as it lies right side up will tend to straighten itself. This seems to be a small point, but it is a very important advantage for filing and for the convenience of those who are to handle the drawings. Also the rough side takes colors and inks better than the glazed side. To trace on the glazed side is not wrong, for it is often done, but it possesses no advantages of its own, and has the disadvantage mentioned above.

Chalk dust scattered over the surface of the cloth after it is tacked down will remove the slightly greasy coating which prevents the ink from flowing well from the pen. This is always necessary if the glazed side be used, and usually for the rough side. The chalk must be carefully removed from the cloth before inking.

The first step in inking is to draw the center lines. Remember that *accurate intersections* are of the utmost importance. No circle is complete without two intersecting lines, preferably at 90 degrees, to determine its center, and these lines should be inked before the circle. When this is done a definite point exists for the needle point of the compasses. If the circle is drawn first the needle point may not be placed accurately at the center on the pencil drawing beneath, and the location be thrown out.

Likewise the principal center lines of pieces, the lines around which the pencil drawing was built up, should be at once put in.

The main body of the drawing, the full lines, should be taken next. In general, circles and arcs should be inked first, but there are cases where it is easier to run the arcs into the straight lines than to match the straight lines to the arcs. They are exceptions. however, and can be judged only as the case arises.

Straight lines, horizontal and vertical, should be inked with the T-square and triangle *in position.* It is a common practice to dispense with the use of the T-square entirely in inking in, using the triangle to match the lines to the arcs already drawn. A necessity for this implies very poor work on the arcs, for with any reasonable care true horizontal and vertical lines will match the arcs all right. With regard to time required, the accuracy with which the T-square may be brought up to a line, or the triangle set on the T-square, more than makes up for the time gained in even an approximate setting of the triangle without a guide. It is just as easy to cultivate the habit of holding the T-square and triangle with the left hand and the pen with the right, and draw an exact line, as to lapse into the other method, which is not workmanlike.

The lines of the body of the drawing depend for their width upon the size of the detail. For a large piece they may be $\frac{1}{32}$ inch wide, and the shade lines $\frac{3}{64}$ inch. For a small detail such widths would be too great. Remember that *contrast* is the principal aim, and to produce it is the only reason why we use different kinds of lines on a drawing. Hence the greatest care must be exercised to prevent body lines from becoming confused with center or dimension lines, and *vice versa.* Also thick lines are desirable for the production of a bold blue-print.

Shade lines are shown on Plates I, K and N only. They are put on according to principles already explained. They certainly improve the drawing from an artistic standpoint, and the student should know how to put them on when desired. Whether or not it is desirable to adopt them on all working drawings is not the purpose of this book to decide, or even discuss. Almost always drawings can be made perfectly clear without them, and are so made and satisfactorily used in probably the majority of shops. Some shops are willing to pay for the extra time necessary to put on shade lines; this, however, is purely their own investment.

Cross-section lines are usually drawn at an angle of 45 degrees with the horizontal, and on sections which are adjacent to each other the slope should be in different directions. If three or more sections come together the width between section lines can be so changed as to indicate clearly the different parts. An example of this is shown in Fig. 6.

The spacing of section lines must not be too fine, rarely closer than $\frac{1}{16}$ inch, more often from $\frac{3}{32}$ to $\frac{1}{8}$ inch, else the labor involved is too great and uniformity practically impossible. It is a waste of time to rule in section lines on the pencil drawing; they may be sketched in freehand, as shown on the original layout of the steam cylinder. Even spacing concerns the tracing alone, and the student should train his eye to regularity as he traces. The thickness of section lines may be intermediate between that of center lines and body lines of the drawing.

Inking Dimensions and Letters. Extension lines may be dotted, as explained in Part III, or they may be fine, full lines, the latter method being illustrated in the series of pump plates in this paper. Dimension lines are also often made fine, full lines. If these lines are made full they should be made as fine as it is possible to draw them and still have them firm, clear lines. The same width should be used as for center lines.

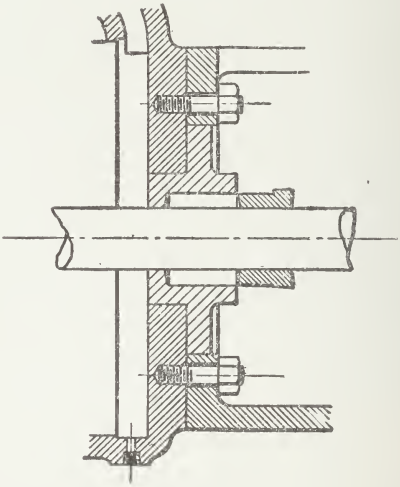


Fig. 6.

Character in inked figures and letters is more difficult to attain than in pencil work. In the first place a pen suitable to the style of drawing is necessary. A civil engineer's fine mapping pen, which gives character to his drawing, is not desirable in producing the bold character of a machine drawing. For the latter choose a rather stiff, blunt pen which is not "scratchy, but runs smoothly, making a line of uniform width. A pen with a round, or ball-shaped nib, recently put on the market, answers the purpose well for ordinary details. A bold, free stroke should be made with the idea of producing a smooth, even line, finished at the first trial. The hesitating uncertainty of the beginner's hand produces a "shaky" letter, and going over a letter or figure twice or more to smooth it up usually makes it worse.

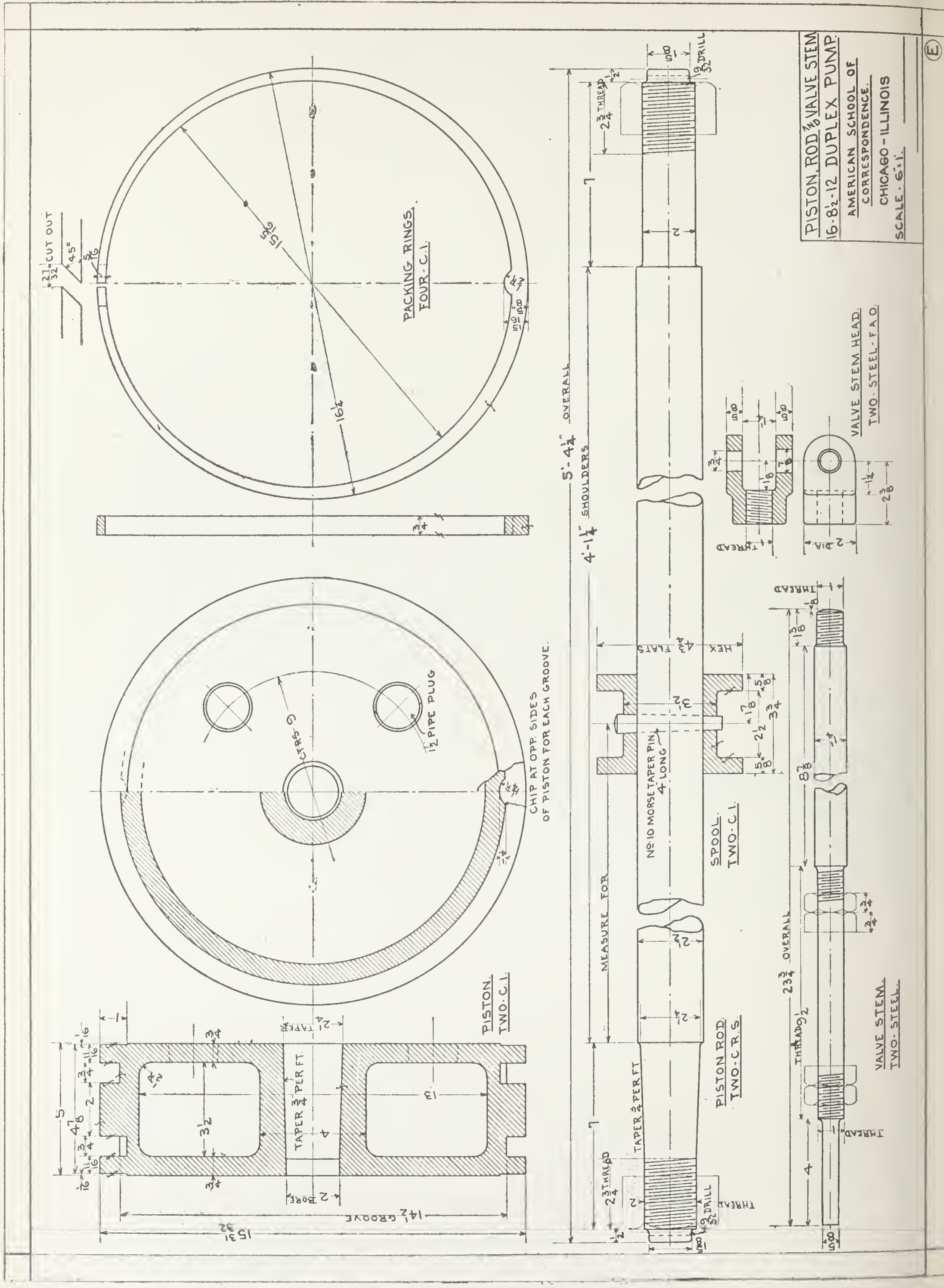
Figures and letters which are broad in proportion to height are easier to make, and have more character. It should never for a moment be forgotten that uniform *height* and *slope* carefully followed will develop character and quickly lead to artistic excellence.

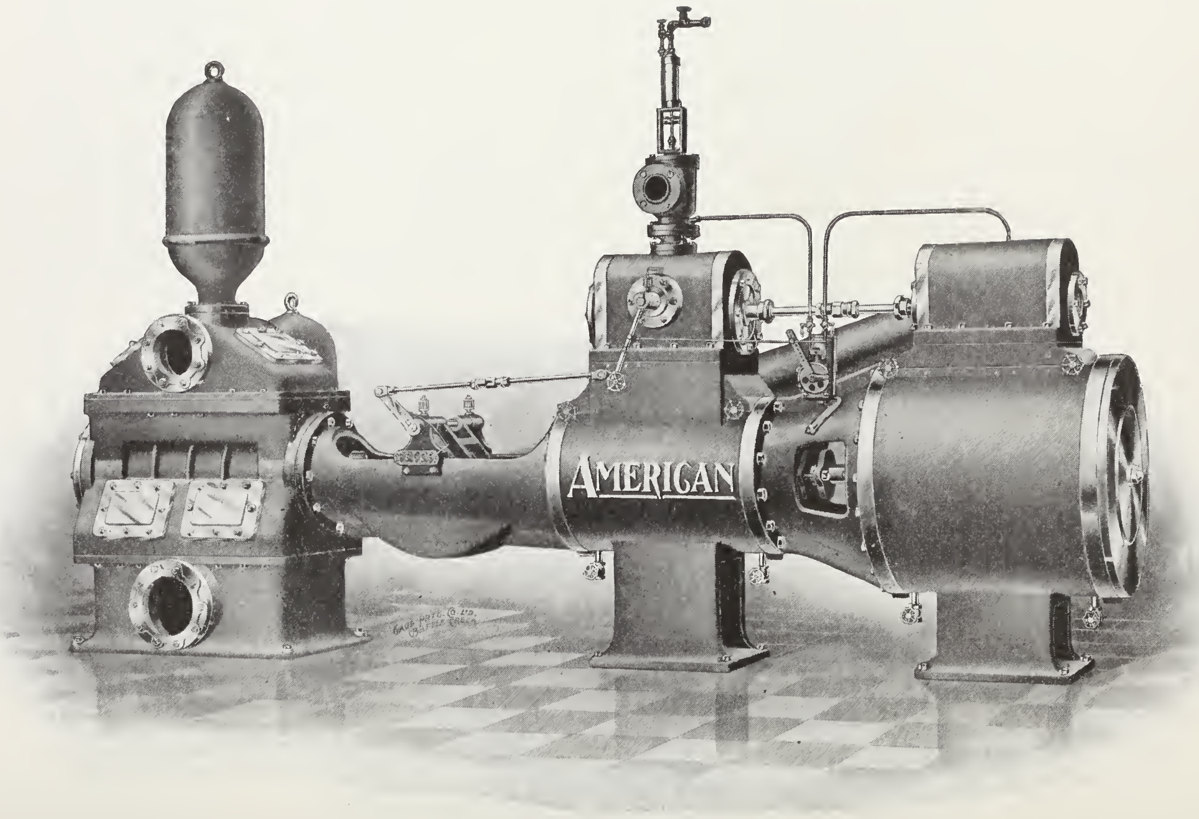
Foot and inch marks are often put after figures according to the common usage. In cases where feet and inches are expressed, thus: 3′ — 6″, or 4′ — 0″, they are, of course, absolutely necessary, and the dash between the figures must be very positively indicated. In cases of inch dimensions alone the marks may be put on if desired, but where there can be no doubt that inches, and not feet, are meant, the inch marks are not necessary. This practice is followed on the plates of this paper.

Abbreviations. A list of the most common abbreviations in use on working drawings follows. This list has been adopted for the plates in Part IV:

F. A. O.	finished all over.
f.	finished surface.
R.	radius.
D.	diameter.
R. H.	right hand.
L. H.	left hand.
P. R.	piston rod.
P. TAP.	pipe tap.
CTRS.	centers.
C. I.	cast iron.
S. C.	steel casting.
Bz.	bronze.
C. R. S.	cold rolled steel.
T. S.	tool steel.
O. H. S.	open hearth steel.
W. I.	wrought iron.

Plate E. Piston Rod and Valve Stem. The piston is of the one-piece box type, with sprung-in rings. The width is reduced to $4\frac{7}{8}$ inches at the outside, so that if the piston strikes the cylinder heads it will not tend to spring and break off the narrow ridge of metal outside of the packing ring. The piston rod is fastened to the piston on a taper drawn in by a nut, and the


PISTON, ROD & VALVE STEM
16-8½-12 DUPLEX PUMP
AMERICAN SCHOOL OF CORRESPONDENCE.
CHICAGO-ILLINOIS
SCALE - 6"=1'.
E
PACKING RINGS. FOUR-C.I.
CUT OUT
45°
15 5/16
16¼
PISTON. TWO-C.I.
TAPER ¾ PER FT.
2¼ TAPER
2 BORE
14½ GROOVE
15 31/32
CTRS-9
1½ PIPE PLUG
CHIP AT OPP. SIDES OF PISTON FOR EACH GROOVE.
5'-4¼" OVERALL
4'-1¼" SHOULDERS
MEASURE FOR
PISTON ROD. TWO-C.R.S.
TAPER ¾ PER FT.
2¾ THREAD
19/32 DRILL
THREAD
HEX 4¾ FLATS
No. 10 MORSE TAPER PIN 4" LONG
SPOOL. TWO-C.I.
VALVE STEM HEAD. TWO-STEEL-F.A.D.
2 DIA
VALVE STEM. TWO-STEEL.
23¾ OVERALL
8⅞
THREAD



THE AMERICAN COMPOUND PUMP.
American Steam Pump Co.

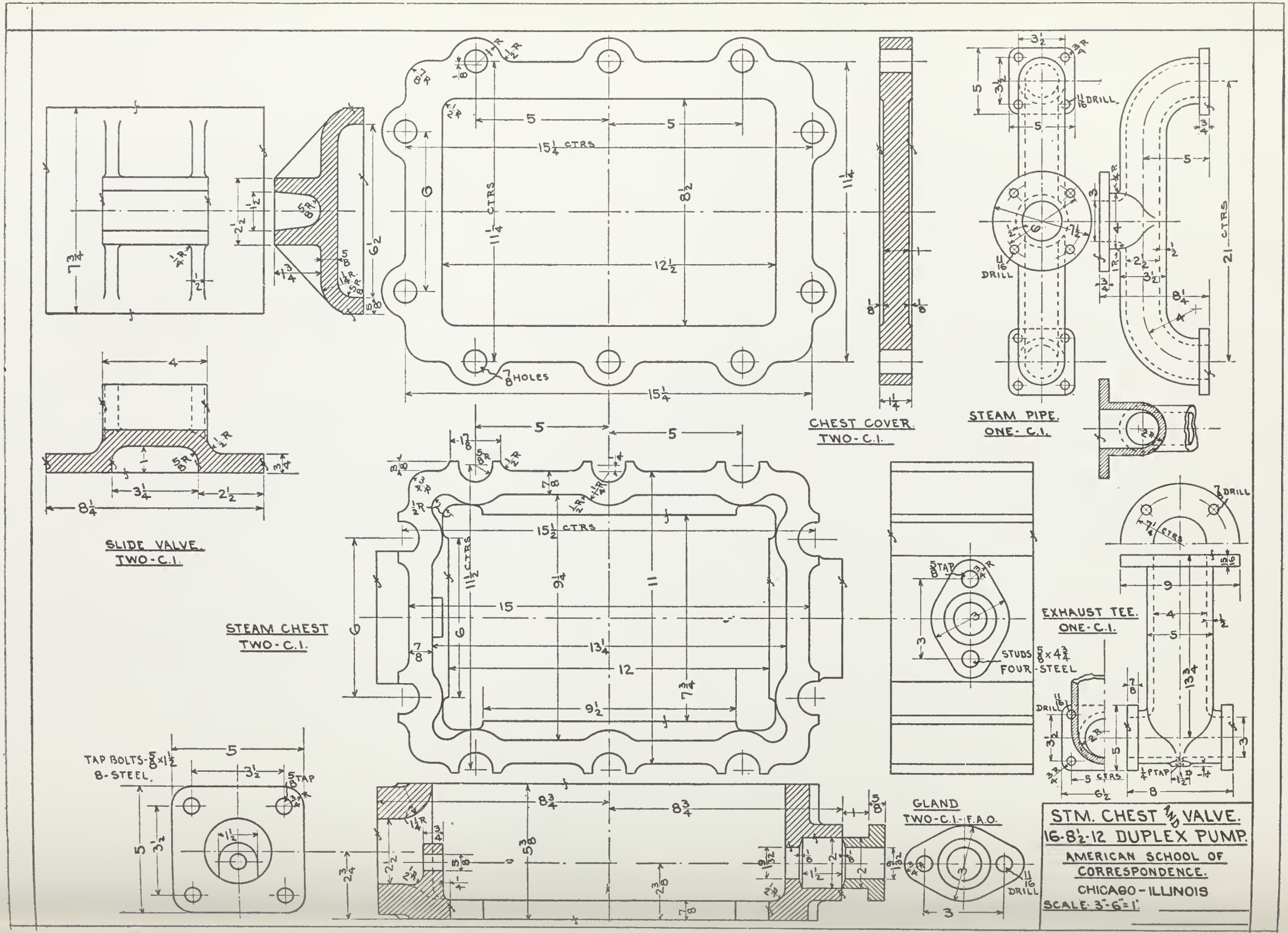
nut is checked by a $\frac{1}{4}$-inch split pin. The packing rings are prevented from slipping round the piston by lugs fitting loosely in chipped recesses in the groove. These being at opposite sides for each groove, the leakage of steam through the split in the ring is minimized, for it must pass half way round the piston before it can pass through the split in the other ring. This is a simple, but fairly effective, device.

The packing rings are usually cast in the form of a cylinder of some length, turned to a diameter a little larger than the cylinder bore, cut off to the required width, and sufficient space cut out to permit being sprung in to the size of cylinder bore.

The location of the spool on the piston rod is not positively known, as the setting of the valve bracket may be slightly different from what the drawing calls for. Hence, instead of a dimension, the words "measure for" are put on, to indicate that the spool be located during the erection of the pump. The hexagonal flanges of the spool are convenient to hold the rod from turning while screwing on the piston and plunger nuts.

Molding and Machining. There are no special features connected with the molding and machining of parts on Plate E. The holes in the piston side walls are necessary to give supports for the core, the piston being cast on its side. These holes, after the core is cleaned out through them, are plugged as indicated.

Plate F. Steam Chest and Valve. The steam chest in this instance is located on the cylinder by fitting down over the ledge made by the valve seat. The side flanges also serve the purpose of guiding the valve. It will be noticed that the steam chest cover is $15\frac{1}{4}$ inches × $11\frac{1}{4}$ inches, while the steam chest is 15 inches × 11 inches. This allows a ledge of $\frac{1}{8}$ inch, all around which the cover overhangs the walls of the chest. The steam cylinder flange in order to correspond must likewise be $15\frac{1}{4}$ inches × $11\frac{1}{4}$ inches. The reason this is done is because of the difficulty of making good matched joints between the cylinder flange, chest and cover. The practice of thus leaving a little ledge all round is by no means universal, and often the irregularity in the joints is smoothed off by chipping. This is the case with the other flanges on this pump. The steam chest, however, was thought less likely to match properly, and the slight overhang gives the finished appearance of a sort of beaded edge.



STM. CHEST AND VALVE.
16-8½-12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO-ILLINOIS
SCALE 3"-6"=1'

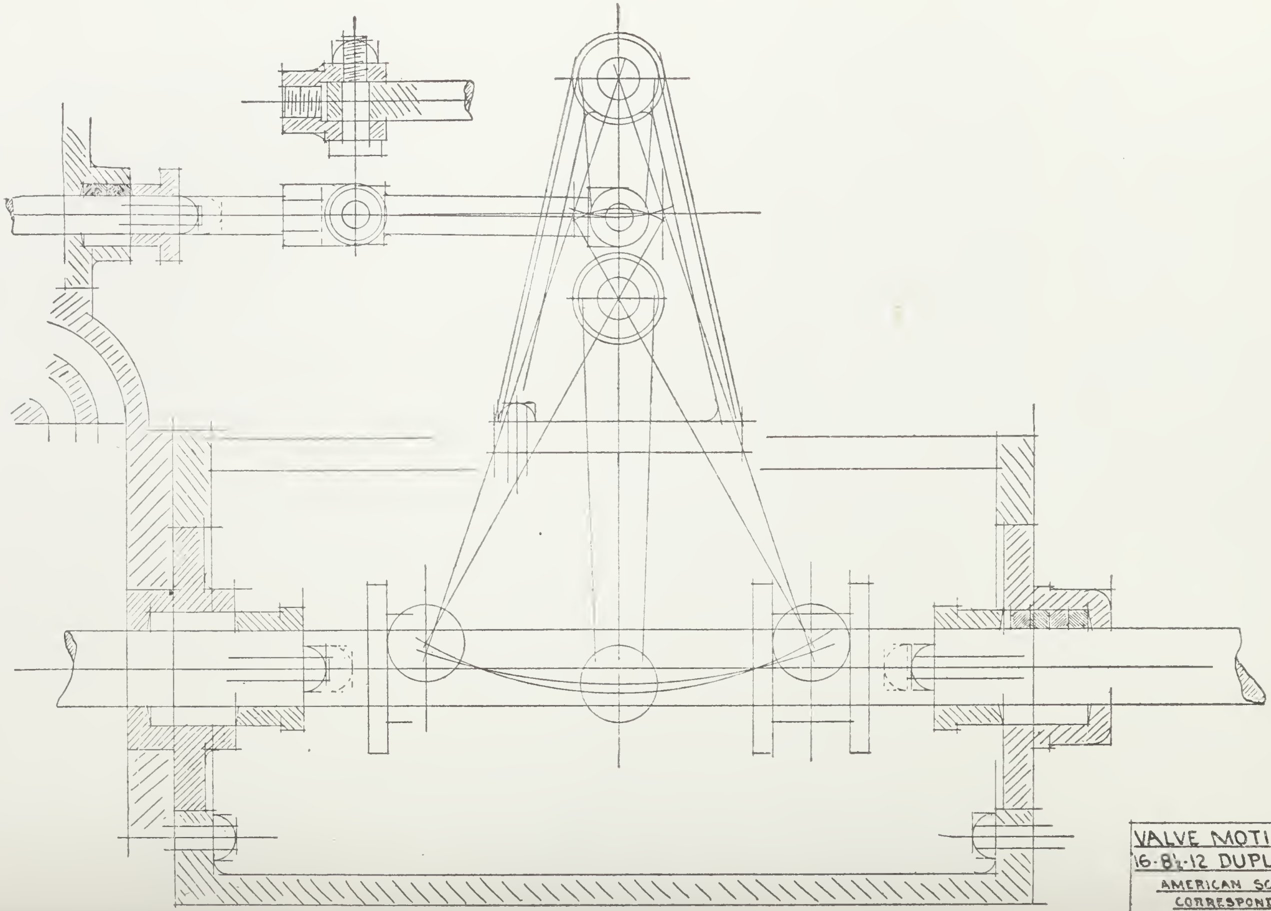
The valve is what is known as a "square" slide valve. This means that when the valve is placed central on the ports its working edges are "square" with the ports; that is, in exact line with them. If the valve be moved either way from this position, the slightest travel will admit steam to one end of the cylinder and exhaust it from the other. (See Plate A.) Another way of stating this is to say that a "square" slide valve is a slide valve without "lap."

The valve is driven from the valve stem by the striking of the nuts against the lug on its top. Since the valve is already guided on its edges by the steam-chest flange, the valve stem, to avoid springing, must be perfectly free in the slot cast for it, as is shown by the $\frac{5}{8}$-inch radius of the bottom, the stem being 1 inch in diameter.

The steam-pipe flange is made square to keep the height of the chest as low as possible. The radius of the bend should be ample; in this case 4 inches is considered sufficient.

The exhaust tee must have its upper flange high enough so that the chest cover can be lifted and slipped off the studs without interfering with it. The lower flanges should be made wide enough to permit the tap bolts to be put in without striking the 4-inch vertical pipe, 5-inch centers being necessary. The $\frac{1}{4}$-inch drip-cock, as located, readily drains the steam chest and exhaust passage of both cylinders, as well as the exhaust tee.

Molding. It is evident that the steam chest will be molded in the position shown on the drawing. The parting line of the mold will be through the centers of the steam-pipe opening and the stuffing-box. These holes must be cored out. The main body of the chest could be made to leave its own core, but it may not be made in this way. It may be cheaper to fashion the pattern solid, and make one large core-box for the inside. In this way the pattern will probably hold its shape better and require less repairs, than if it were made in green sand. The core-box will be an extra piece to make, but it probably will cost no more than to carve out the inside of the pattern, and is a rather more substantial job when done. The molding can be satisfactorily done by either method, shop conditions being the controlling element. As far as the labor of molding alone is concerned, the first method is probably easier, as it saves handling large cores.



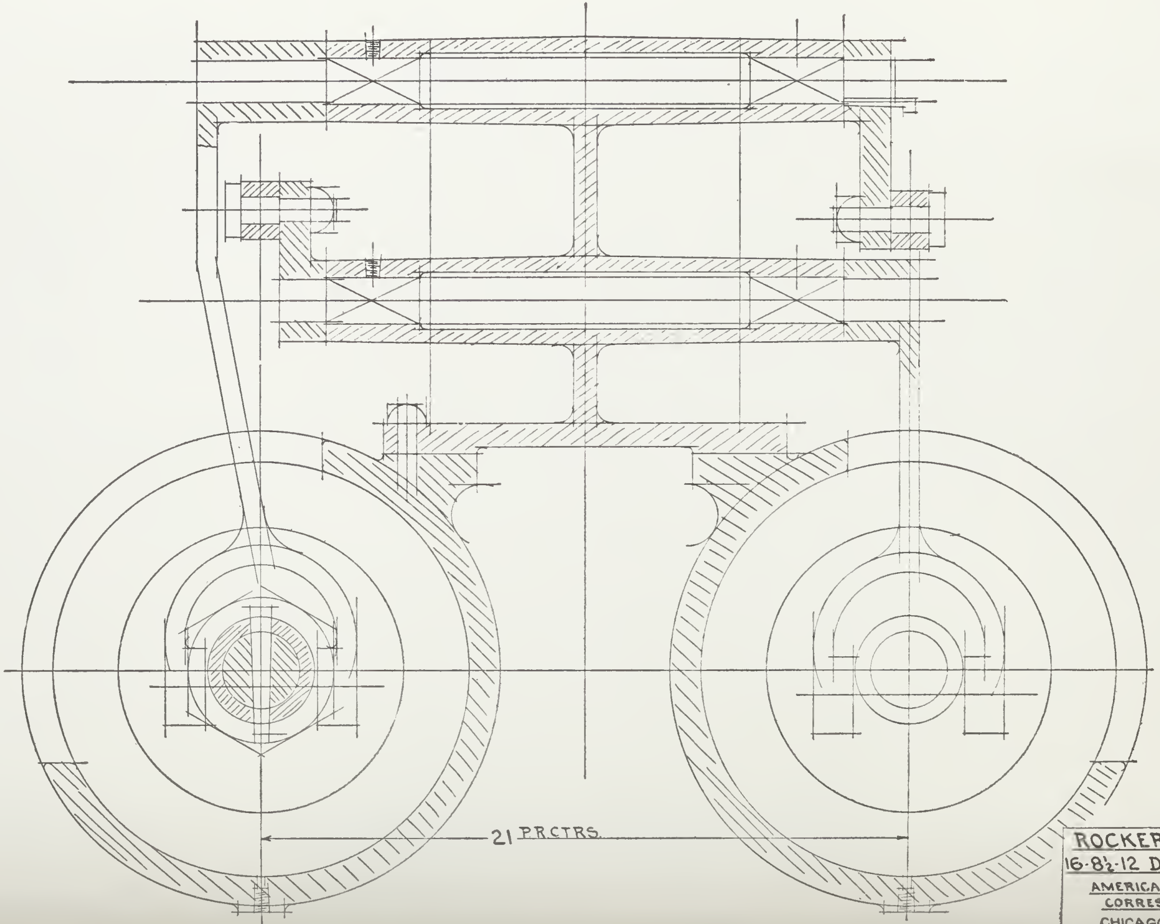
VALVE MOTION LAYOUT
16·8½·12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE · 6"-1'.

The other parts in Plate F are very simple in their molding, and require no special attention.

Machining. Most of the surface work on this plate is adapted to the planer. The slide valve may, perhaps, if finished in lots of considerable number, be more satisfactorily handled on the milling machine. The final finish of the face of the valve must be a scraped fit to its seat.

The drilling of the cover and pipe flanges is to actual layout on the casting, or preferably, through jig plates. A templet for laying out is at least desirable, even though the expense of a jig plate be not deemed necessary.

Plates C and D. Valve Motion Layout. These plates represent the layout of the valve motion, and are necessary in order to find the length of the levers and rocker arms. It will be noticed in Plate D that the valve stem of one side of the pump is controlled by the movement of the piston-rod of the other side, the proper direction of motion being given to the valve by placing the rocker shaft above or below the valve stem as required. By reference to Plate A it will be further noticed that the nuts on the valve stem inside the chest, which abut against the faces of the lug on the valve, do not rest against the faces of the lug in the position shown, but have considerable lost motion. This lost motion is one of the essential features of the valve motion of a duplex pump, and permits the valve to remain at rest for a short period at the end of the stroke, though the valve stem may have reversed its motion and begun its return stroke. When this lost motion is taken up by the movement of the stem and the nuts abut against the lug on the valve, the valve will move, and from this point to the end of the stroke be positively controlled by the motion of the stem. At the end of the stroke the stem will reverse, when the lost motion will again permit the valve to rest for the same period as at the other end, and then move on as before. The time of rest of the valve, and consequently the pistons and plungers, is approximately one-third the period of the stroke. This means that the piston on one side travels one-third of its stroke before it picks up, through the valve levers, the valve on the other side. During the second third of its travel it is bringing the



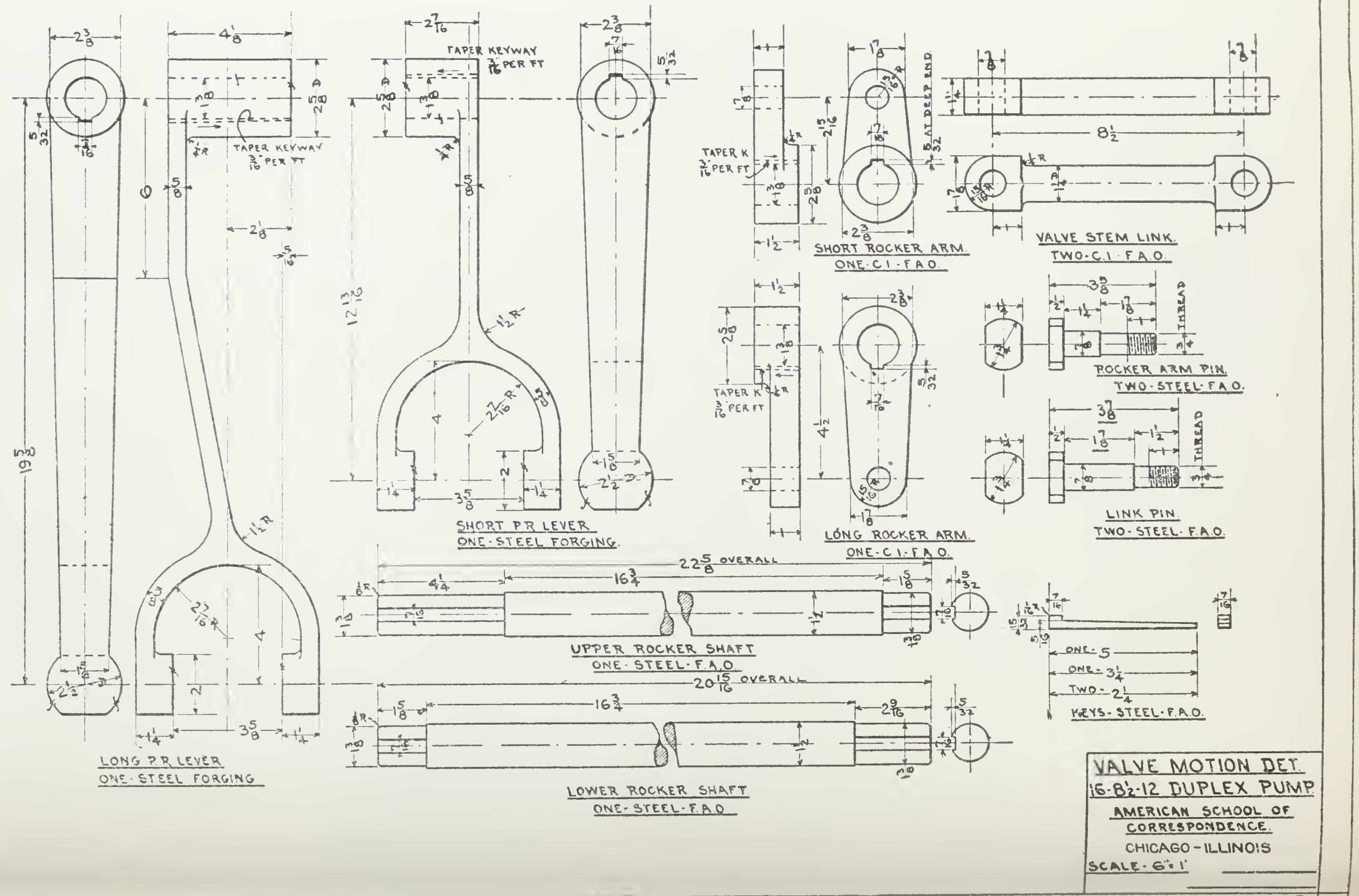
ROCKER ARM LAYOUT.
16-8½-12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE - 6"=1'.

valve to the point of opening. During the last third of its travel it is opening the port, wider and wider, to steam. Thus the opposite piston will start when the first piston has covered two-thirds of its stroke, and there will be only one-third of the stroke when both pistons are moving at the same time.

This relative period of rest to motion is not always made in this exact ratio, but is at least approximate to it. The period of rest at the end of the stroke is to allow the water end to adjust itself quietly to the reversal of motion about to take place at the end of the stroke. When the plunger stops, the water valves must be given time to seat themselves, and the flow of water through the passages checked. It is much easier to start the flow in the opposite direction if the reversal of plunger motion is not instantaneous. Hence for handling long columns of water, which, once in motion, tend by considerable energy to remain in motion, the duplex pump by this peculiar delayed action has been found to be well suited.

It will be found that for complete uncovering of port, and motion divisible into thirds as described, the travel of the valve stem should be three times the width of port, or $3 \times \frac{7}{8}$ inch $= 2\frac{5}{8}$ inches. A little more than this is allowed, and the travel made $2\frac{7}{8}$ inches in this case. Referring to Plate C, this distance is laid off as shown by the two limiting vertical lines across the line of the valve stem, the central vertical line of mid-position being drawn. The problem then is to find such centers for the rocker arms that the travel of the piston-rod spool will, through proper leverage, produce travel of the valve stem between these two vertical lines. This can readily be done by a few trials, the only requirement for this case being that the extremes of the arc of swing of both piston-rod lever and rocker arm shall be equally above and below the center of piston rod and valve stem respectively. The greatest possible travel of the piston-rod spool, $12\frac{1}{2}$ inches, is usually laid out in this case, not the nominal 12 inches.

From this layout the lengths of the levers and arms may be scaled off for the detail drawing, also the location of the rocker-arm centers. The student has the former given him on Plate K, but the latter, which is necessary for the development of Plate L, must be determined by his own layout. Plate D must also be



VALVE MOTION DET.
16-8½-12 DUPLEX PUMP
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE · 6" = 1'

Ⓚ

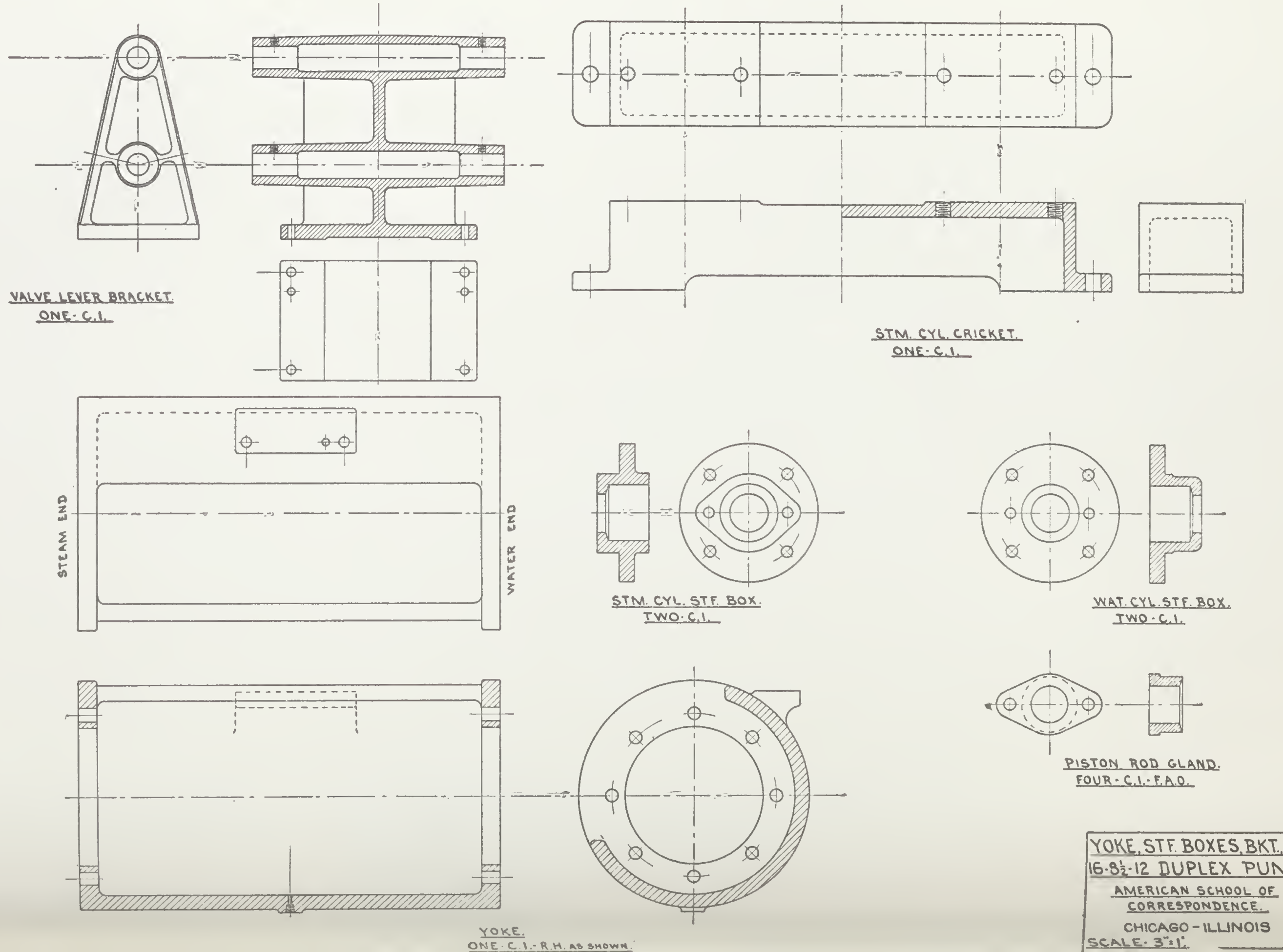
laid out before developing the cross-section of the valve bracket.

The design of stuffing boxes for both steam and water ends, and the length of the yoke, should be determined next. A safe method of assuming clearance between the spool and the gland studs at the end of the stroke, is to imagine that the gland stud nuts have accidentally worked off the studs, so that they are about to drop. They are thus shown by dotted lines on Plate C. A good clearance, say $\frac{1}{4}$ inch to $\frac{1}{2}$ inch, is then allowed, and the gland drawn in. The length of the gland is determined by the number of rings of packing necessary in the stuffing box; it is usually provided that the gland may compress the packing to about one-half its original depth before bringing up against the face of the box. Packing $\frac{5}{8}$ inch square will do for this size of piston rod, hence the faces of the yoke are easily determined, and its detail, with the stuffing boxes, proceeded with as on Plate L. The length of yoke may be brought to an even figure; and proceeding on the above plan the length can be conveniently made in even inches without any fractions; viz., 28 inches.

It will be noticed that the stuffing-box flanges serve to center the yoke in line with the steam and water cylinders. This is a desirable feature of construction, and forms a simple and easy method for lining up the steam and water ends.

Plate K. Valve Motion Details. The piston-rod levers on this plate are specified to be steel forgings. Forgings of this kind are expensive, but are light, neat and reliable for the important service which they have to perform. Castings, whether steel or iron, are much cheaper, and perhaps more commonly used for this detail. When sound they are equally serviceable, though of more clumsy proportions; but the danger in castings of this form is the existence of hidden flaws or pockets, which frequently occur at the points where the hub or the fork joins the arm. These flaws cannot be readily detected from the outside, and breakage may occur at some critical time, when the disability of the pump may be a serious matter.

The use of shade lines is illustrated on this plate. The increased artistic effect is noticeable, but it would seem that absolute clearness would still exist, even if shade lines were not used.



YOKE, STF. BOXES, BKT., ETC.
16-$8\frac{1}{2}$-12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE - 3" = 1'.

It will be noticed that on the detail of the "link pin" two of the dimensions have a short "wavy" line beneath the figures. This is one of the several ways of indicating that the dimension is "out of scale." Some draftsmen use a straight dash beneath the figure; some draw a circle about it; some print after the figure, "out of scale." Although workmen are not allowed to scale drawings, but are required to "work to figures only," yet for general safety's sake, and for the sake of the draftsmen who consult the drawings frequently, attention must be called to any variation of the figure from the measured distance on the drawing. Nothing makes a workman, or any one else who reads a shop drawing, lose confidence in it more quickly than to discover that it does not "scale"; but when no indication exists that the draftsman himself is aware of it, then every dimension is viewed with doubt and hesitation, and the drawing becomes practically worthless.

Dimensions seldom should be out of scale; but if they are, through error or necessary change, a carefully worded note should be added.

Molding and Machining. No special features of molding or machining are noteworthy on Plate K.

Plate L. Yoke, Stuffing-boxes, Bracket, etc. Having worked up the layouts of Plates C and D, the student has enough information to proceed with Plate L. This, like Plate G, is without dimensions, the student's work being to make the drawing and fill in the necessary shop data.

The valve-lever bracket is bolted down to its lug on the yoke through holes larger than the bolt, thus permitting slight adjustment. When the proper location is determined, the bracket is positively fixed in position by two dowels, $\frac{1}{2}$ inch in diameter. The holes in both bracket and yoke are drilled through both pieces at the same operation. This very common method of fixing bolted parts of machinery in absolute position not only assures firmness, but also in case of removal, permits the part to be readily and positively replaced in its exact original position.

If possible, the steam cylinder cricket should be of such height that the stone or brick work upon which it rests shall be at the same level as that beneath the water cylinder. The tapped holes in the top surface receive bolts from the cylinder foot. These bolts are often used only for shipping purposes, the cylinder foot when the pump is set up being allowed to slide freely on the cricket, thus permitting free expansion and contraction. In such cases the water end is rigidly fastened to the foundation by holding down bolts.

Molding and Machining. The valve lever-bracket would most naturally be molded with the axes of the shafts vertical, the parting line of the mold being the center line of the middle web. This makes quite a long "draw" for the shaft bosses, but the ample taper on the outside overcomes this difficulty. The space between the side webs leaves its own core. The shaft cores stand on end in the mold, which is the best position for strength and stability.

Another method is to have the parting line of the mold on the vertical center line of the bracket, as shown in the end view. In this case the bracket would be cast on its side, and cores must be set for each side of the middle web. The shaft cores are set as easily as before, but in this case lie flat. As with the steam chest, each method has its advantages, which depend largely upon existing conditions. As cored work is generally avoided whenever possible, the first method would probably be chosen.

The shaft bosses are "chamber-cored," to save labor in boring, the bearing surface for the shaft being only a short distance at the ends. The chamber-core diameter should be enough larger than the shaft so that by no possibility can the cutter run into the rough scale, even if the hole be bored slightly out of line. If it should do this, the labor of caring for the cutters more than offsets the attempted saving of labor.

The yoke is simply a barrel open at each end, and with a piece cut out of its side. The inside evidently must be cored out, and the core is satisfactorily supported at the ends on its horizontal axis. The parting line of the mold may be either the vertical or horizontal axis of the end view, the only difference being that in one case the ledge for the valve bracket will "draw," and in the other case it must be loose on the pattern and "pulled in" after the main pattern is drawn.

The cricket and stuffing boxes present no difficulties. The bore of the stuffing boxes and glands should be from $\frac{1}{32}$ inch to $\frac{1}{16}$ inch larger than the rod, to allow the fit to be entirely between the rod and the packing.

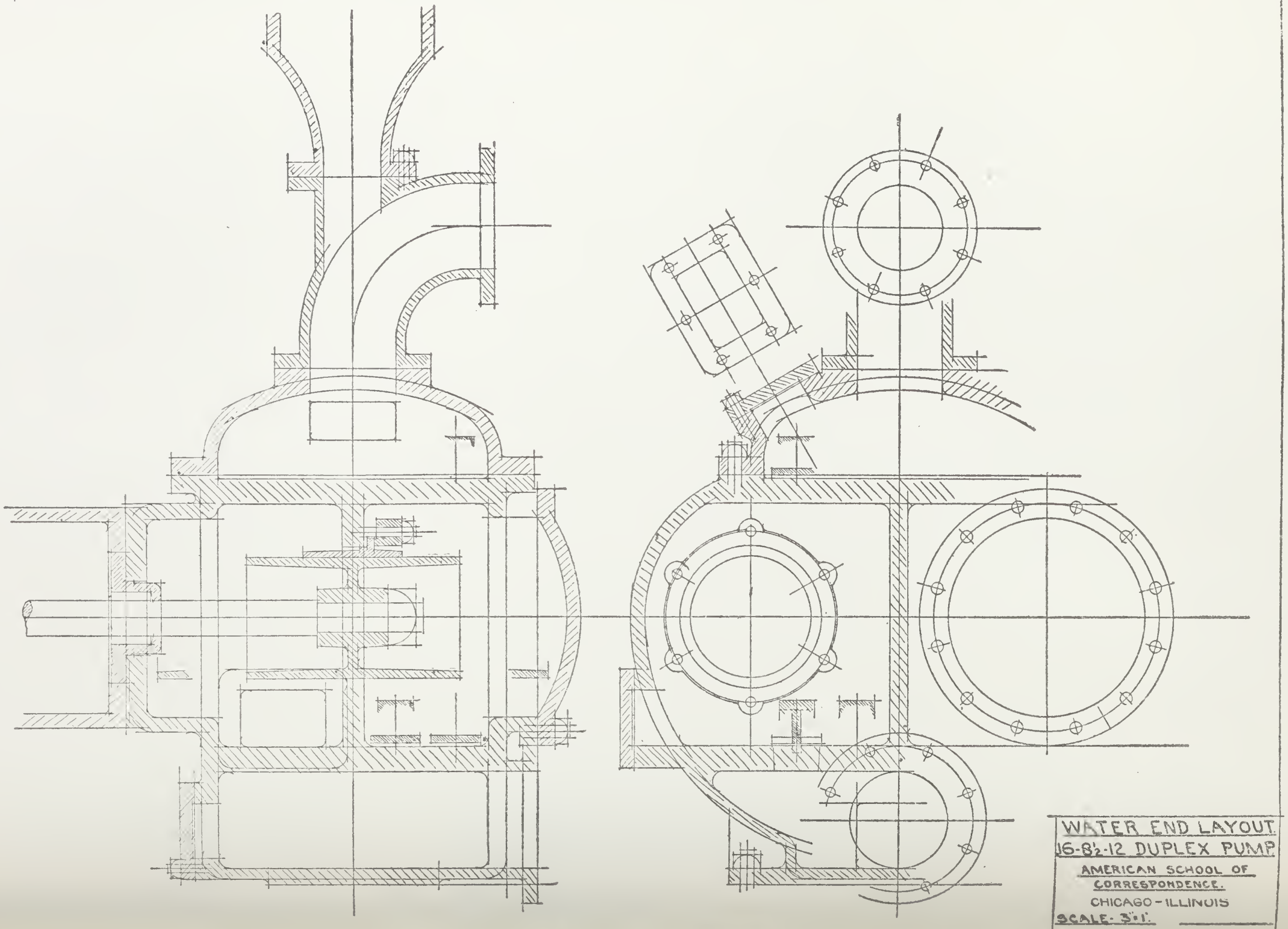
The horizontal boring machine with a double facing head is adapted to boring and facing the yoke flanges. The drilling is accomplished as before by templet or jig.

Attention is called to the tapped holes for oil or grease cups on the valve-lever bracket. The holes on the lower boss cannot be drilled strictly as shown, because the drill shank will not clear the upper boss. They should be swung around the boss at such an angle as will allow the drill to clear. This is a good instance of the common error of drawing details which cannot be made, and constant watch must be kept to avoid such mistakes.

Plate B. Water End Layout. As in the preceding work, Plates H and I being given in full detail offer a good start for the development of the water cylinder, which is the purpose of Plate B. As before, work should begin at the inside and progress outwards. Thus the piston rod with its nut should be drawn first, the hub of the plunger built around it, then the plunger barrel, the bushing, and ring to clamp the bushing. The limits of the plunger travel should be sketched in, and the valve outline shown in order to determine clearances. The progress of Plate B is on exactly the same basis as that stated in detail for the steam cylinder layout; hence it need not be repeated.

The points controlling the design of the water end must, however, be studied to enable the student to work intelligently. The fit of the rod into the plunger hub is loose, $\frac{1}{16}$-inch play being allowed, in order to permit the plunger to be guided solely by its bushing, and thus be independent of any change of alignment of the piston rod.

The relative length of plunger and bushing should allow the end of the plunger to overrun the edge of the bushing at the termination of the stroke, to prevent the formation of a shoulder. The bushing is made of brass because of the better bearing of the two dissimilar metals, brass and iron. Of course there is no lubrication except the water, and the dissimilar metals tend to "cut" less than if both were alike. The brass bushing also prevents the plunger from "rusting in" in case of long periods of



WATER END LAYOUT.
16-8½-12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE - 3"=1'.

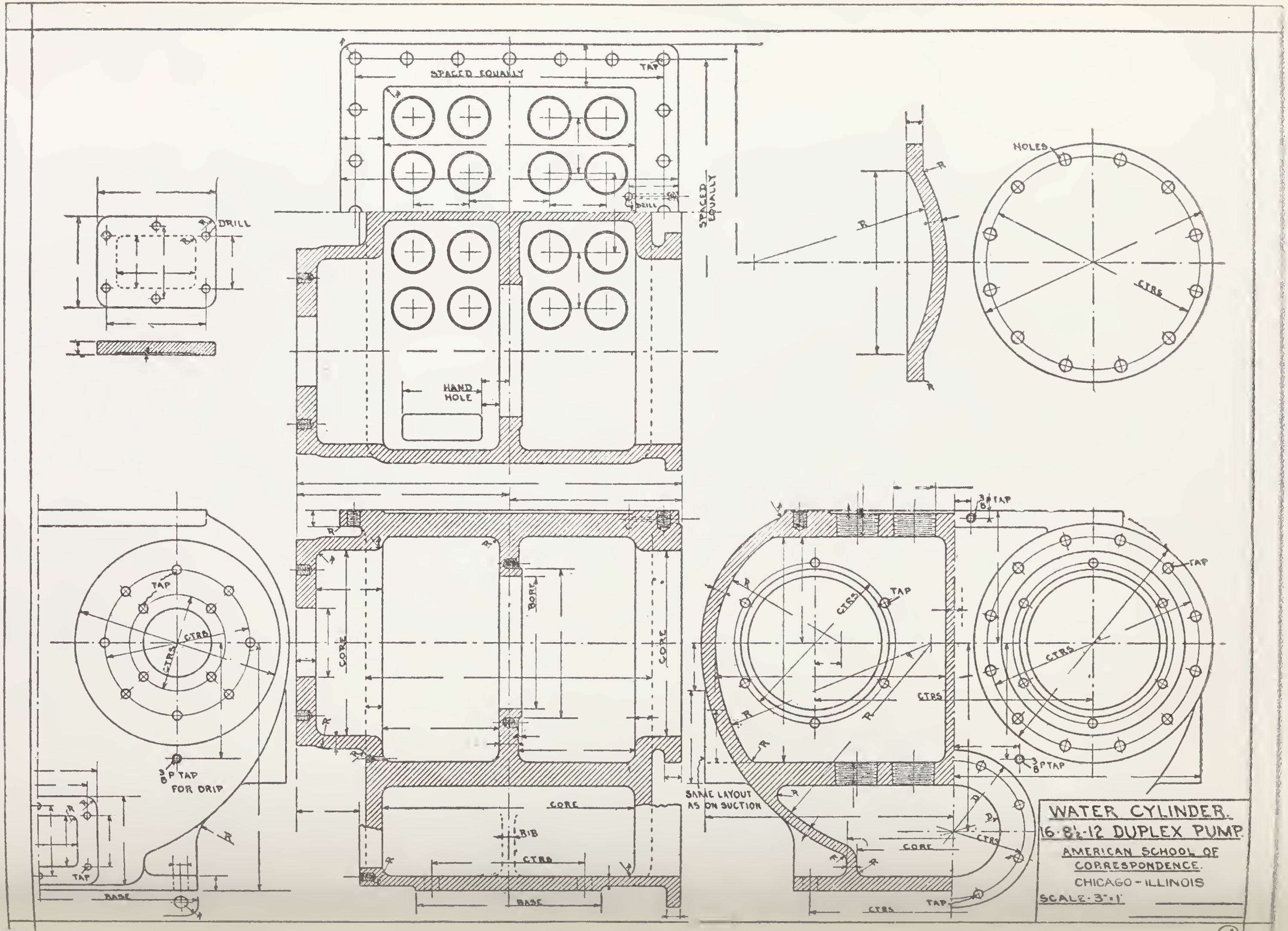
disuse. The bushing being of expensive material is made as light as possible, hence it has no stiffness of its own. Therefore it is reinforced by a deep cast-iron ring, which also takes the bolts and clamps the bushing tightly to its ground seat. These stud bolts are usually made of "tobin bronze," a rust-proof material, possessing strength almost as great as that of steel. This arrangement permits ready removal of the bushing when necessary.

As the parts of the common pump valve illustrated in detail on Plate H must be often replaced during service of the pump, provision must be made for unscrewing the stem and substituting a new one. This must be done through the hand holes provided on the cylinder. The lower valve deck must be located so that the inner valves when unscrewed will not strike the clamp ring. As shown in Plate B the clearance is pretty small, almost too small, but as it affects only two valves, it will probably cause no inconvenience. No hand holes are necessary for the end chambers, as access to the valves is had by removing the outer heads. The upper deck may be placed at a height giving sufficient clearance to allow the upper nuts of the clamp ring to be unscrewed with a socket wrench from the end of the pump. These decks are subjected to a severe pounding from the pulsations of the pump, and should be amply strong; $1\frac{3}{4}$ inches is deemed thick enough for this case.

The middle transverse wall may be $1\frac{1}{2}$ inches thick and the middle longitudinal wall a little thinner, about $1\frac{1}{4}$ inches. With high pressures these walls, being flat surfaces and the valve decks likewise, are likely to fracture under the heavy pounding. To avoid making them excessively heavy they are often strongly ribbed, either on the inside or outside, usually the former.

The curving side walls are of better form to withstand pressure, and need not be as thick, 1 inch being sufficient. This can be decreased to $\frac{3}{4}$ inch in the suction passage below the deck, where little pressure exists.

The outer head is also considered strong enough at 1 inch thickness, on account of its curved shape. It requires $\frac{7}{8}$-inch studs. Studs are preferred to tap bolts in this case, as in all other similar cases, on account of the frequent unscrewing of the nuts for purpose of removal. One or two unscrewings of a tap bolt in cast


WATER CYLINDER.
16·8½·12 DUPLEX PUMP.
AMERICAN SCHOOL OF CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE · 3" = 1'

iron will destroy the tightness of the thread, while the stud, being steel, stands the wear better.

The valve seats are taper screwed into the deck; they are sometimes forced in on a plain taper fit. They are located as closely as strength of the deck between the holes will permit. It is not well to place the edge of the valve closer than $\frac{1}{2}$ inch from the cylinder walls. The valve holes in the lower deck should be in line, or nearly so, with the holes in the upper deck, in order to allow the shank of the mill to pass through when milling the lower holes.

The suction opening is 7 inches in diameter, $12\frac{1}{2}$-inch flange, $10\frac{1}{2}$-inch bolt circle, $\frac{3}{4}$-inch tapped holes.

By means of the hand hole at the end of the suction passage, any dirt which may have been brought in through the suction pipe may be removed.

The water cylinder cap, discharge ell and air chamber may be laid out from the detail Plate I, and the student must do this to see that the parts actually go together properly.

With the foregoing discussion the student should be able to produce Plate B, which is the preliminary step to the detail drawing of the water cylinder as shown on Plate J.

Plate J. Water Cylinder. The water cylinder is, perhaps, the most complicated detail that the student will meet in this set of plates. Fundamentally, it is simply a box with curved sides, divided by the several walls into five compartments, each of which communicates with the outside by a round nozzle or flange. If this basic idea be kept constantly in mind, the student will have no trouble in building up the detailed design.

This fundamental conception of a complicated piece is a very important idea, and should be developed carefully by the student. It is one of the great secrets of good design, both from an artistic and a commercial standpoint. We often see a machine which seems to begin anywhere and end nowhere; it appears to be a miscellaneous collection of bosses, lugs, ribs and flanges. There is no general prevailing shape to the structure, no harmony of the lines. This is because the designer, if he may be so called, did not have the fundamental notion of shape, to which all minor details should have been subordinated. He simply grouped parts together, without considering the fundamental structure.

In this water cylinder the box is the basic part of the structure, and its lines must be first developed; they should be designed to convey a smooth, regular and consistent surface to the eye. Then the nozzles and flanges may be added as subordinate parts; they will merely interrupt, but not destroy, the prevailing outline of the box. The dotted lines in the cross-section views of Plate J show the general shape behind and beneath the nozzles.

The hand holes are the same as on Plate I, and the detail of the cover should specify the number required for both places.

Provision for draining the four chambers of the water cylinder is made by the $\frac{3}{8}$-inch pipe tap holes at the lower deck, and the cap, likewise, by the single hole at the upper deck. Drip cocks are screwed into these holes.

The holding-down bolts should not be less than 1 inch diameter; $1\frac{1}{4}$ inch would perhaps be better; and the holes in the foot should be drilled at least $\frac{1}{8}$ inch large.

Dimensions. It will be noticed that this plate has dimension lines, but no figures. This is because the cylinder is rather difficult to figure, and it is desired to guide the student in arrangement of the figures without lessening the benefit of his study of them. Special attention should be paid to this feature of the plate. Notice that although space for dimensions is restricted, a clear opening is always found for the figures; and when one view seems to offer no space for a figure, another view gives the desired opportunity.

No finish marks or titles are shown on this plate, these being left entirely to the student for insertion.

Molding. The centers of the curves for the sides being on the main horizontal axis of the nozzles, the cylinder, if molded to be cast vertically as shown, will draw readily both ways from this line. The exceptions to this easy draw are the foot, suction nozzle and flange, and hand-hole boss. On account of the inside of the cylinder being cored, these pieces if made loose on the pattern have ample space to be "pulled in" after the main pattern is withdrawn.

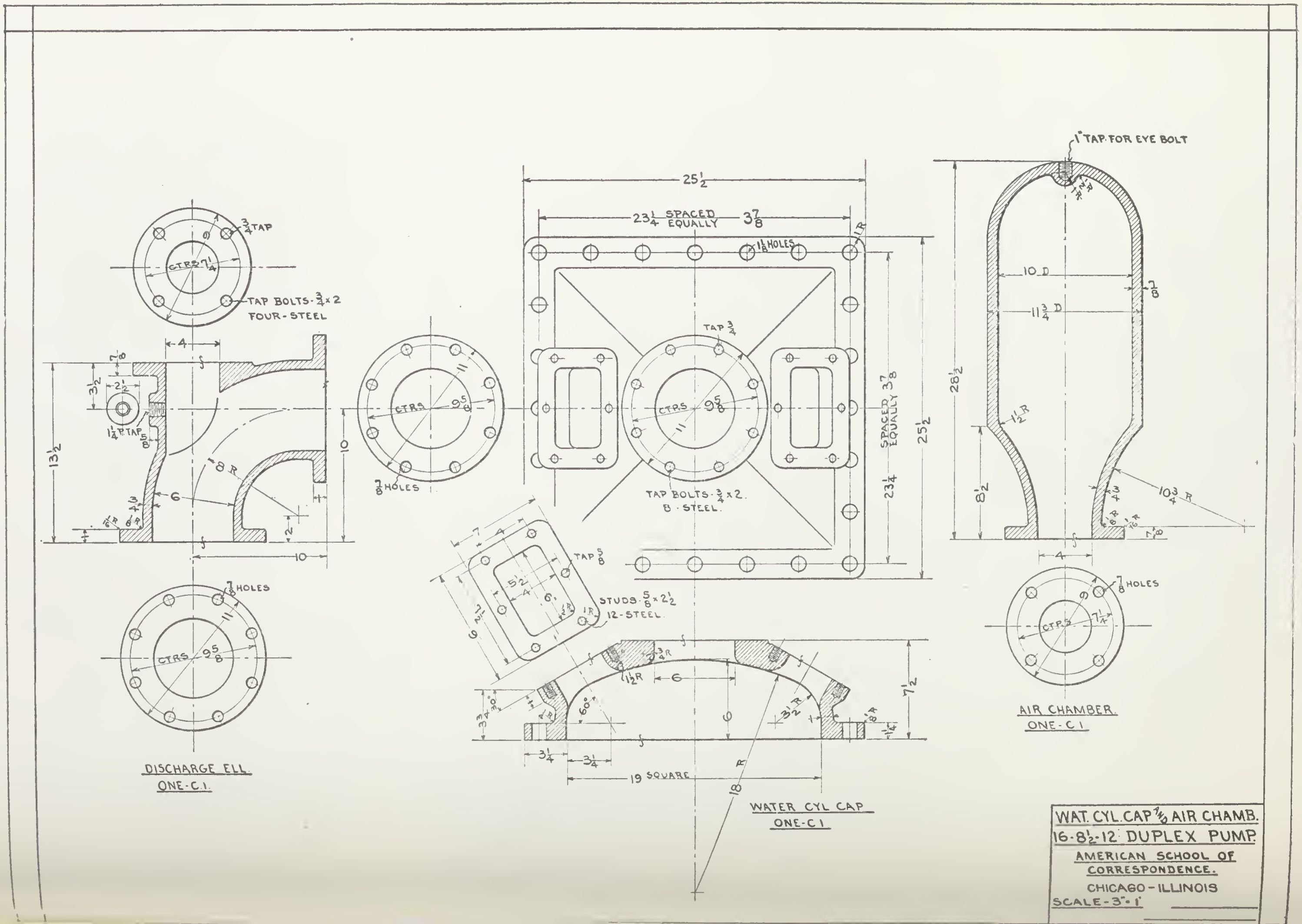
The suction passage below the deck communicates with the main core through the valve holes, hence it may be supported from the main core. This involves some difficulty, however. If a three-part flask be used, and another parting established at the center of the suction flange, in addition to the previous one, the problem becomes much simplified.

It is desirable to make the four chambers of the cylinder alike in general proportions. It is then possible to make a single core-box, and by the use of loose pieces change the length of the nozzle cores and transpose from right to left, thus saving labor on the pattern. This, however, multiplies the loose pieces on the pattern. The many pieces are likely to become lost and make frequent repair necessary. Hence it is not always wise to use a single core box too much, and good judgment is required to fix the limit.

Machining. Special double horizontal boring machines are now in common use for such cases as this water cylinder. The centers are made adjustable, so that within limits any distance between piston-rod centers can be met. The advantages of double boring are, of course, most obvious for a considerable number of duplicate cylinders.

It will be noticed that the face of the suction flange is carried out flush with the cylinder head face. This affords opportunity for finishing all the end surfaces at a single setting of the tool, whether the work be done on the rotary or reciprocating planer. This same point might have been observed on the small hand-hole boss at the other end of the cylinder, but the advantage gained did not seem to warrant extending the " reach " through the hand hole.

Plate I. Water Cylinder, Cap and Air Chamber. For a water cylinder cap of this size, the most difficult problem is to find room for the hand hole bosses. A hand hole 4 inches × 6 inches is about as small as can be used, and this calls for a flange at least 7 inches × 9 inches. These are the proportions shown on the plate, and since the boss overhangs the bolts in the main-cap flange, it must be cut away underneath to clear the nuts. If three stud bolts are used on each side, this overhang also requires that the nut be " fed on " ; that is, screwed on little by little as the end of the stud protrudes above the flange when the cap is being lowered into place. This is an awkward process, but is sometimes necessary.



WAT. CYL. CAP & AIR CHAMB.
16-8½-12 DUPLEX PUMP
AMERICAN SCHOOL OF CORRESPONDENCE.
CHICAGO-ILLINOIS
SCALE-3"-1'

The discharge ell should have an easy bend; usually the radius is somewhat more than the outside diameter of the pipe, in this case 50 per cent greater. It is customary on this piece to provide an opening for the attachment of a relief valve as shown, $1\frac{1}{4}$-inch pipe tap. This valve can be set to open at a desired pressure, so that the water end may be relieved in case of accidental excessive pressure.

The air chamber provides an air cushion for the water to make the delivery more constant, and take the shock which would otherwise come with hammer-like force and full intensity upon the cylinder. Being placed at the highest point of the water end, air will naturally tend to collect in the air chamber and keep it charged. In some cases, however, a special charging device is necessary.

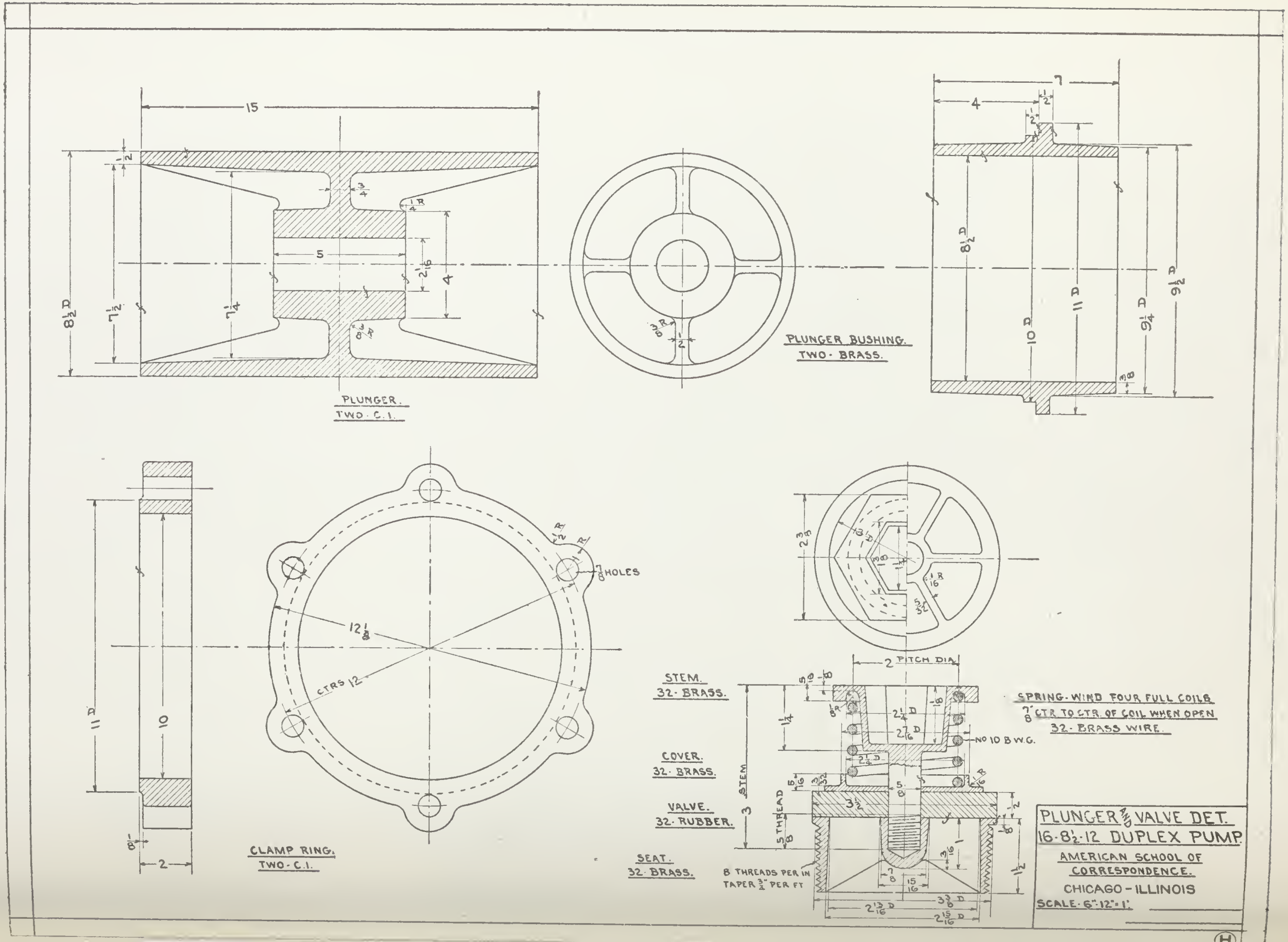
Molding and Machining. The hand holes being at an angle will not "draw." Hence cores must be set for these openings at least, and it may be desirable to core out the whole inside of the cap for the sake of keeping the pattern in good shape by making it solid. Otherwise it is easy to let it leave its own core.

The overhang of the hand-hole bosses requires loose pieces for the overhanging part. They are "pulled" in after the pattern is drawn.

The molding and machining which are further required on details of Plate I are simple, and require no special discussion.

Plate H. Plunger and Valve Details. This plate is noticeable for illustrating a method of drawing details not used elsewhere in this set of plates. On the other plates each piece is separately detailed. On Plate H the details of the valve, cover, seat, stem and spring are shown assembled, and dimensioned without separation. This is an allowable method when clearness is not sacrificed, but it is usually found desirable only with simple construction. It concentrates parts on the drawing, and probably saves some time, besides showing the workman just how the parts go together. The only test which the student need to apply in this, as in any method of detailing, is the test for absolute clearness.

It is believed in the case of the valve as shown that the details are completely illustrated without sacrificing clearness. Special care in putting in dimensions is of necessity required.



PLUNGER AND VALVE DET.
16 - 8½ - 12 DUPLEX PUMP.
AMERICAN SCHOOL OF CORRESPONDENCE.
CHICAGO - ILLINOIS
SCALE - 6" = 12" = 1'.

The valve stem can be unscrewed either with a socket wrench on the inside or an ordinary fork wrench on the outside.

The seat, after being screwed to position in the deck, is often faced off, to true up any distortion caused by screwing in.

The valve itself, of rubber, can be bought of any desired grade of hardness. The specification for any given set of valves depend upon the quality of the water, the pressure and the general service of the pump.

Molding and Machining. By reason of the simple nature of the parts on this plate, the molding and machining is left entirely to the original consideration of the student.

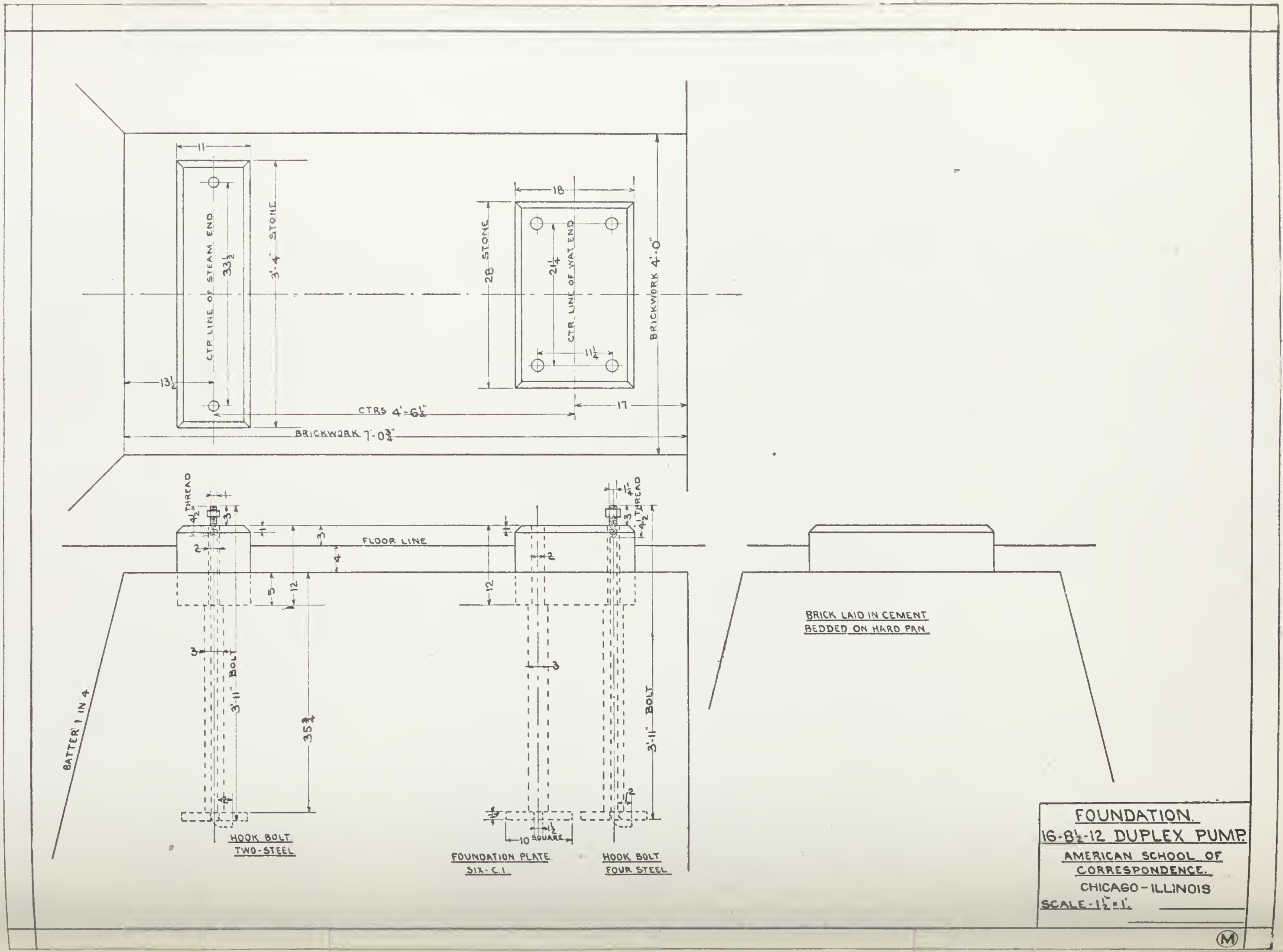
Plate M. Foundation. Pumps are often set directly upon a foundation of brick, but it makes a better job to bed stones, with surfaces dressed plane and true, into the main foundation, and rest the pump feet upon these stones. The simplest form of holding down bolts are shown on Plate M, a plain hook at the lower end, pulling up against a flat cast-iron plate, to distribute the pressure into the brickwork. These plates are of course bedded, and the bolts set as the foundation is built up. As the subsequent courses are laid some little space is left around the bolts, which may be afterwards filled with cement, thus making the bolts rigid with the foundation.

The water end of the foundation has no batter, because the suction pipe often drops vertically down from the end of the pump, and clearance is therefore necessary.

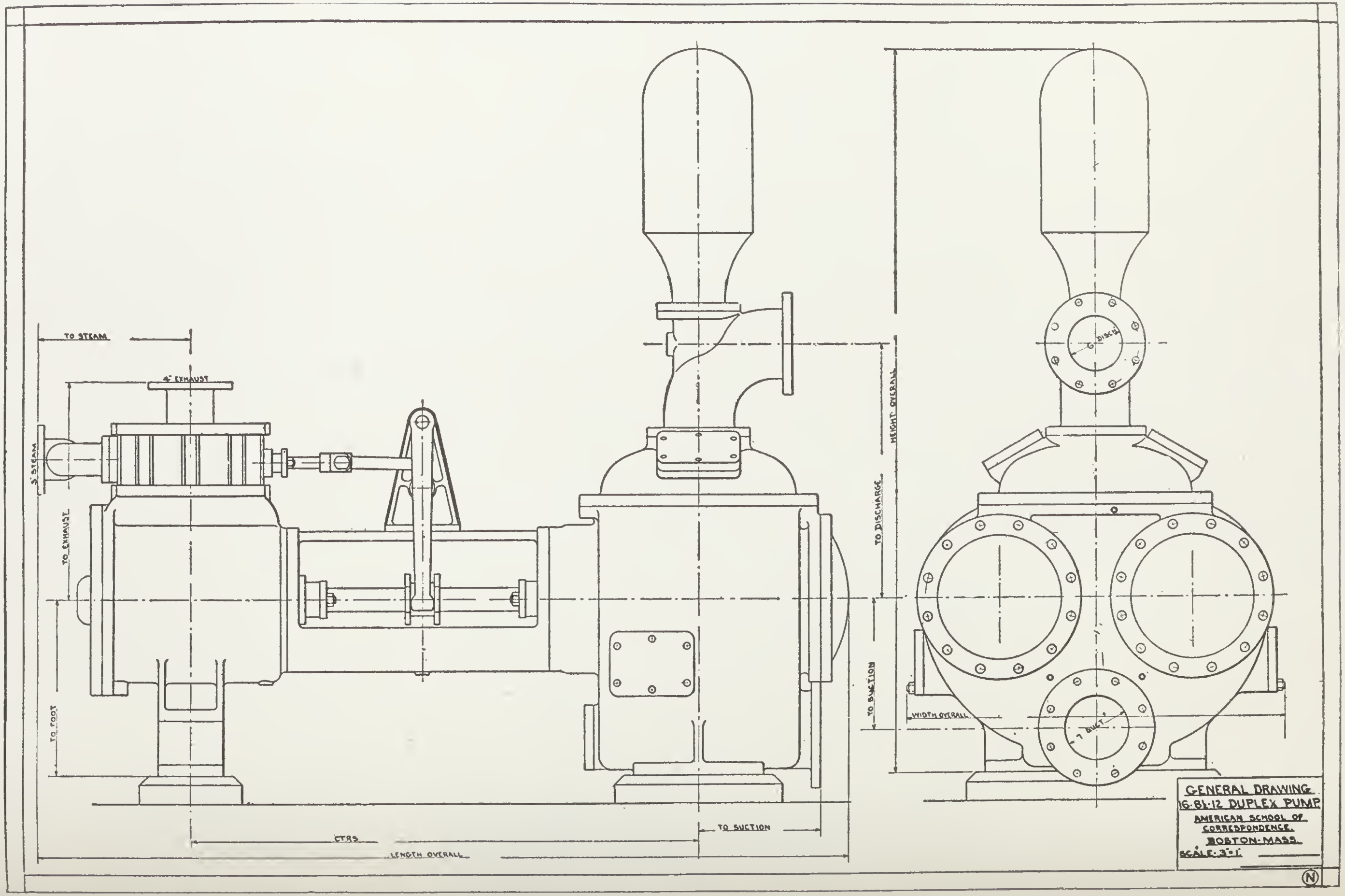
The floor line is placed 4 inches above the brickwork, to allow for the usual 1-inch top floor and 2-inch plank beneath, and still have a space left for shims to level the floor.

Plate N. General Drawing. This is an example of a plain, everyday shop drawing, to show the relation of parts and the extreme space occupied by the pump. A great deal of time can be needlessly wasted in producing a drawing of this character, by trying to make too faithful a picture. For example: If all the bolt heads were put in, it is safe to say that several hours' extra time would be required for this one item alone. But the drawing would be no better for shop use. Hence all bolt heads and nuts have been left out, except when necessary to show clearance.

Shade lines have been put on for no special reason except



FOUNDATION.
16-8½-12 DUPLEX PUMP.
AMERICAN SCHOOL OF
CORRESPONDENCE.
CHICAGO-ILLINOIS
SCALE-1½"=1'.


TO STEAM
4" EXHAUST
3" STEAM
TO EXHAUST
TO FOOT
CTRS
LENGTH OVERALL
TO SUCTION
TO DISCHARGE
HEIGHT OVERALL
TO SUCTION
WIDTH OVERALL
6" DISCH.
7" SUCT.
GENERAL DRAWING
16·8½·12 DUPLEX PUMP
AMERICAN SCHOOL OF
CORRESPONDENCE.
BOSTON-MASS.
SCALE 3"=1'
N

that if they are desirable on any drawing they are especially desirable on a general drawing, where one part overlaps another, as they make it easier to pick out and separate one surface from another. Some lines are shaded in this drawing which are not strictly sharp edges. It is held, however, that the rounding of a corner ought not to destroy its character as an edge casting a shadow, and such lines are treated accordingly.

An assembled or general drawing of this character should be laid out strictly from the dimensions shown by the details. It thus serves a valuable purpose in checking up figures, and showing whether or not the parts will go together. The method or character of the work in no respect differs from that suggested for the detail drawings.

If a scale of 3 inches = 1 foot be used, the size of sheet must be 24 inches × 36 inches. The student, however, will perhaps find it easier to use a scale of $1\frac{1}{2}$ inches = 1 foot, in which case the ordinary size, 18 inches × 24 inches, will suffice. For such a small scale it will be found undesirable to attempt to put in any very small fillets and corners, although those that can be readily handled by the ordinary bow pen ought not to be omitted. As a matter of fact, the expert draftsman either leaves the corners sharp, as suggested, or puts in the smallest curves freehand.

Order Sheets. Any set of drawings is incomplete unless in connection with it a statement is made in tabular form of the complete make-up of the machine. An infinite variety of ways exists for making the specifications. Sometimes the tabulated data are placed on the general drawing. Most often, however, printed blanks are provided, usually of bond paper, arranged with special reference to the individual shop system and methods of handling work; these blanks are filled in by the draftsman, indexed and filed as a part of the set of drawings. They can be blue printed for use in the shops the same as a drawing. From these sheets stock is ordered, checked off, and watched in its process of manufacture.

Order sheets are indispensable in any well-ordered shop. Hence they are illustrated on pages 59, 60, 61 and 62 as the final step in the set of pump drawings. They are made as simple as possible, and are not intended to fit any special shop system. As previously stated, the exact form and method of classification can be determined only when the shop conditions are known.

The student, having carefully followed through the preceding pages, must not think that he is master of pump construction, for even the type illustrated has been but touched upon. The object of the detailed discussion is to get the student in close touch with the spirit of construction, to make his drawings real serious work. It is hoped that the student will work just as though a machine were to be built from his drawings, and built to sell at a profit. Only in this way can advanced work in mechanical drawing be of benefit to him, for after becoming expert in the use of the instruments, no other advance is possible except advance in *thought.*

DATE.
MAY 20, 1902.

AMERICAN SCHOOL OF CORRESPONDENCE
AT
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

TYPE.
INSIDE PLUNGER.

LIST OF CASTINGS

FOR

16–8½–12 DUPLEX PUMP.

No. Wanted.	Name.	Drawing No.	Patt. or Piece No.	Material.	Remarks.
2	Steam Cylinder	G		C. I.	R. & L.
2	Steam Cylinder Head	G		C. I.	
2	Steam Chest	F		C. I.	
2	Steam Chest Cover	F		C. I.	
2	Slide Valve	F		C. I.	
1	Steam Pipe	F		C. I.	
1	Exhaust Tee	F		C. I.	
2	Valve Steam Gland	F		C. I.	
2	Piston	E		C. I.	
8	Piston Pipe Plug, 1½″	E		C. I.	
4	Piston Packing Ring	E		C. I.	
2	Spool	E		C. I.	
1	Steam Cylinder Cricket	L		C. I.	
2	Steam Cylinder Stuffing Box	L		C. I.	
2	Water Cylinder Stuffing Box	L		C. I.	
4	Piston Rod Gland	L		C. I.	
1	Valve Lever Bracket	L		C. I.	
2	Yoke	L		C. I.	R. & L.
1	Short Rocker Arm	K		C. I.	
1	Long Rocker Arm	K		C. I.	
2	Valve Stem Link	K		C. I.	
1	Water Cylinder	J		C. I.	
2	Water Cylinder Head	J		C. I.	
3	Hand Hole Cover	J		C. I.	
1	Water Cylinder Cap	I		C. I.	
1	Air Chamber	I		C. I.	
1	Discharge Ell	I		C. I.	
2	Plunger	H		C. I.	
2	Plunger Bushing	H		Brass	
2	Clamp Ring	H		C. I.	
32	Valve Stem	H		Brass	
32	Valve Cover	H		Brass	
32	Valve Seat	H		Brass	
6	Foundation Plate	M		C. I.	

DATE.
MAY 20, 1902.

AMERICAN SCHOOL OF CORRESPONDENCE
AT
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

TYPE.
INSIDE PLUNGER.

LIST OF STEEL AND MISCELLANEOUS PARTS
FOR
16—8½—12 DUPLEX PUMP.

No. Wanted.	Name.	Drawing No.	Patt. or Piece No.	Material.	Remarks.
2	Valve Steam Head	E		St.	Drop Forging.
2	Piston Rod	E		C. R. S.	
2	Valve Stem	E		St.	
1	Long P. R. Lever	K		St.	Forging
1	Short P. R. Lever	K		St.	Forging
1	Upper Rocker Shaft	K		St.	
1	Lower Rocker Shaft	K		St.	
2	Rocker Arm Pin	K		St.	
2	Link Pin	K		St.	
1	Long P. R. Lever Key	K		St.	Drop Forging
1	Short P. R. Lever Key	K		St.	Drop Forging
2	Rocker Arm Key	K		St.	Drop Forging
32	Valve Spring	H		Brass wire	Spring Temper
32	Valve	H		Rubber	Medium

DATE.
MAY 20, 1902.

AMERICAN SCHOOL OF CORRESPONDENCE
AT
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

TYPE.
INSIDE PLUNGER.

LIST OF BOLTS, NUTS AND PINS

FOR

16—8½—12 DUPLEX PUMP.

No. wanted.	Name.	Drawing No.	Patt. or Piece No.	Material.	Remarks.
24	Cylinder Head Stud $\frac{7}{8}$ x $3\frac{1}{4}$	G		St.	
20	Steam Chest Stud $\frac{3}{4}$ x $8\frac{3}{4}$	G		St.	
4	Valve Stem Gland Stud $\frac{5}{8}$ x $4\frac{3}{4}$	F		St.	
8	Piston Rod Gland Stud $\frac{3}{4}$ x 4	L		St.	
24	Water Cylinder Head Stud $\frac{7}{8}$ x $3\frac{1}{4}$	J		St.	
12	Clamp Ring Stud $\frac{3}{4}$ x $4\frac{1}{2}$	J		Tobin bz.	
24	Water Cylinder Cap Stud 1 x $3\frac{1}{2}$	J		St.	
18	Hand Hole Cover Stud $\frac{5}{8}$ x $2\frac{1}{2}$	J		St.	
12	Hand Hole Cover Stud $\frac{5}{8}$ x $2\frac{1}{2}$	I		St.	
8	Exhaust Tee Tap Bolt $\frac{5}{8}$ x $1\frac{3}{4}$	G		St.	
16	Yoke Tap Bolt $\frac{3}{4}$ x 2	G		St.	
8	Steam Cyl. Stf. Box Tap Bolt $\frac{3}{4}$ x $1\frac{3}{4}$	G		St.	
8	Steam Pipe Tap Bolt $\frac{5}{8}$ x $1\frac{1}{2}$	F		St.	
4	Valve Lever Bracket Tap Bolt $\frac{5}{8}$ x $1\frac{3}{4}$	F		St.	
4	Steam Cyl. Cricket Tap Bolt 1 x $2\frac{1}{4}$	F		St.	
16	Yoke Tap Bolt $\frac{3}{4}$ x 2	J		St.	
8	Water Cyl. Stf. Box Tap Bolt $\frac{3}{4}$ x $1\frac{3}{4}$	J		St.	
8	Discharge Ell Tap Bolt $\frac{3}{4}$ x 2	I		St.	
4	Air Chamber Tap Bolt $\frac{3}{4}$ x 2	I		St.	
2	Hook Bolt (special) 1 x 3′—11″	M		St.	
4	Hook Bolt (special) $1\frac{1}{4}$ x 3′—11″	M		St.	
1	Eye Bolt Standard 1″	I		St.	
34	Standard Nut $\frac{5}{8}$			St.	
44	Standard Nut $\frac{3}{4}$			St.	
36	Standard Nut $\frac{7}{8}$			St.	
26	Standard Nut 1			St.	
4	Standard Nut $1\frac{1}{4}$			St.	
4	Standard Nut 2			St.	
8	Special Valve Stem Nut 1			St.	$\frac{3}{4}$ Thick
4	Piston Rod Split Pin $\frac{1}{4}$ x 2	E		St.	
2	Spool Taper Pin No. 10 Morse Taper	E		St.	4″ long
4	Valve Bracket Dowel Pin $\frac{1}{2}$ x 2	L		St.	

DATE.
MAY 20, 1902.

AMERICAN SCHOOL OF CORRESPONDENCE
AT
ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL.

TYPE.
INSIDE PLUNGER.

LIST OF SPECIAL FITTINGS, WRENCHES, ETC.

FOR

16—8½—12 DUPLEX PUMP.

No. Wanted.	Name.	Drawing No.	Patt. or Piece No.	Material.	Remarks.
4	Drip Cock ½″	G			
1	Drip Cock ¼″	F			
2	Drip Cock ¼″	L			
4	Oil Cup ¼″	L			
5	Drip Cock ⅜″	J			
1	Relief Valve 1¼″	I			175 lbs. pressure
1	Standard Fork Wrench ⅝″				
1	Standard Fork Wrench ¾″				
1	Standard Fork Wrench ⅞″				
1	Standard Fork Wrench 1″				
1	Socket Wrench ¾″				12″ handle
1	Valve Stem Fork Wrench				
1	Valve Stem Socket Wrench				

PLATES.

The plates of Part VI (A to N, inclusive,) are so arranged and described that the complete set may be made by the student. In case of insufficient time, the following short examination is prescribed. The student in any case should read carefully all the text, and follow the discussion of all the plates, whether or not actually produced by him.

PLATE XXV.

Reproduce Plate E (page 32), putting on shade lines. Size of plate to be 18 inches × 24 inches, trimmed, with border $\frac{1}{2}$ inch from edge; size within border line, 17 inches × 23 inches.

PLATE XXVI.

Make steam end layout, Plate A (page 10). This is to be done accurately *in pencil.* Size same as Plate XXV.

PLATE XXVII.

Make drawing of steam cylinder, Plate G (page 18), complete, with finish marks and dimensions. Do not put on shade lines.

PLATE XXVIII.

Draw valve motion layout, Plate C (page 36). This plate also is to be done accurately *in pencil.*

PLATE XXIX.

Make rough freehand sketches in pencil of parts on Plate L (page 42), putting on finish marks and dimensions. Use a medium pencil, and make sheet of regular size (18 inches × 24 inches). Arrangement to be similar to Plate L. Do not try to make the sketches to scale, but make them sufficiently complete so that detail drawings can be made from them.

PLATE XXX.

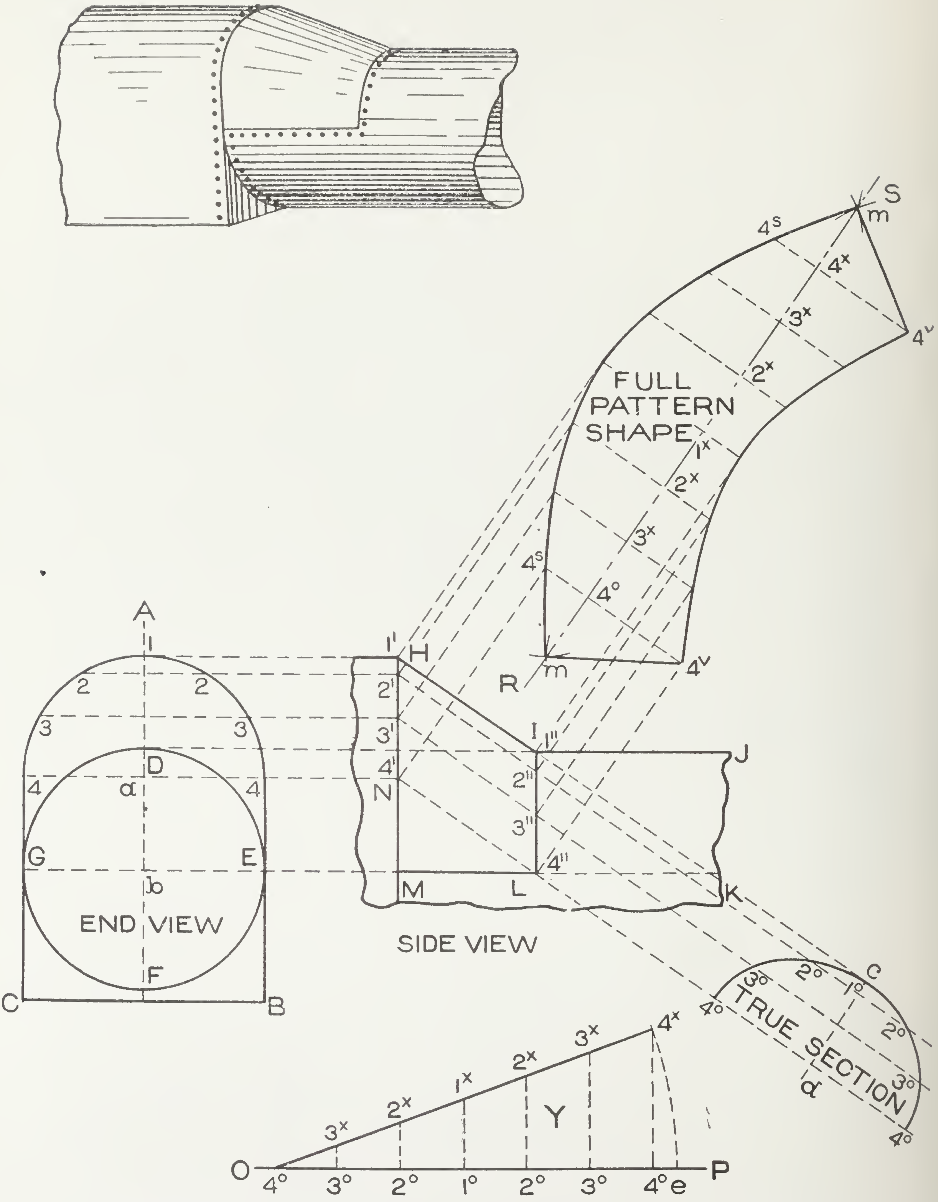
Make a tracing on tracing cloth of Plate E (page 32).

PLATE XXXI.

Make general drawing Plate N. Use scale of $1\frac{1}{2}'' = 1'$, or $3'' = 1'$.

PLATE XXXII (Optional).

Make drawing of water cylinder, Plate J (page 48).



PATTERN FOR GUSSET SHEET OF BOILER, SHOWING DEVELOPMENT.

SHEET-METAL WORK.

PART I.

The sheet-metal worker of today who wishes to succeed must know far more than was necessary years ago. There are many good, practical sheet-metal workers in the trade who are handicapped because they are unable to lay out the patterns that arise in their daily work. Notwithstanding the introduction of labor-saving machinery, the demand for good workmen has increased. While most sheet-metal workers acquire practical knowledge in the shop, they lack the technical education necessary to enable them to become proficient as pattern cutters and draftsmen. In this course, special attention is given to the fundamental principles that underlie the art and science of pattern drafting.

Practical workshop problems will be presented, such as arise in everyday practice, thus giving the student the practical experience that usually comes only after long association with the trade.

CONSTRUCTION.

In constructing the various articles made from sheet metal, various gauges or thicknesses of metal are used. For all gauges from No. 20 to No. 30 inclusive, we assume in the development of the pattern, that we are dealing with no thickness, and we make no allowance for bending or rolling in the machine. But where the metal is of heavier gauge than No. 20, allowance must be made for shrinkage of the metal in the bending and rolling operations, which will be explained in connection with development in heavy sheet-metal work. What has been said about wiring, seaming, and transferring patterns in the Tinsmith's Course is applicable to this course also. It is sometimes the case that the capacity of a vessel or article must be determined, when the rules given in Mensuration should be followed. When figuring on sheet-metal work, the specifications sometimes call for various metals, such as galvanized sheet iron or steel, planished iron, heavy boiler plate, band iron, square or round rods for bracing, etc., zinc, copper, or brass; and the weight of the metal must often be calculated together with that of stiffening rods, braces, etc. On this account it is necessary to have tables which can be consulted for the various weights.

TABLES.

There is a wide difference between gauges in use, which is very annoying to those who use sheet metal rolled by different firms according to the various gauges adopted. It would be well to do away with gauge numbers, and use the micrometer caliper shown in Fig. 1, which determines the thickness of the metal by the decimal or fractional parts of an inch.

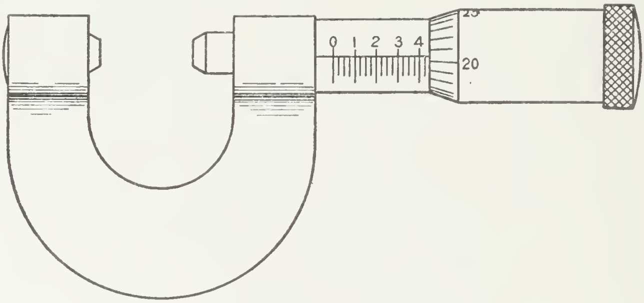


Fig. 1.

This is the most satisfactory method for the average mechanic who works sheet metal manufactured by firms using different gauges. The tables on pages 61 to 74 can be consulted when occasion arises.

SHOP TOOLS.

In allowing edges for seaming and wiring, we must bear in mind that when a seam is to be grooved by hand or machine the allowance to be made to the pattern should conform to the rolls in the machine or the hand tools in use. The edges of the pattern are usually bent on the sheet-iron folder, or brake, while the seam can be seamed or grooved with the hand groover or giant grooving machine. Where round pipe work is done in lengths up to 3 feet, the slip roll former is used, while square or rectangular pipes are bent up on the brake in 8-foot lengths. Where pipes, elbows, stove bodies, furnace shells, metal drums, etc., are made, the sheets are cut square on the large squaring shears, rolled, grooved, and stiffened, by beading both ends in the beading machine, using ogee rolls. There is also a special machine for seaming the cross seams in furnace pipes, also a set of machines for the manufacture of elbows used in sheet-metal work. As before mentioned, if these machines are at hand, it will be well to make slight modifications in the patterns so that both the machines and patterns may work to advantage.

PATTERNS OBTAINED BY VARIOUS METHODS.

In this course will be explained the four methods used in developing patterns for sheet-metal work, namely, parallel line, radial line, triangulation, and approximate developments. What was said on parallel and radial line developments in the Tinsmith's Course is applicable to this course also.

INTERSECTIONS AND DEVELOPMENTS.

The following problems on parallel line developments have been selected because they have a particular bearing on pipe work arising in the sheet-metal trade. All of the problems that will follow should be carefully studied, drawn on cheap paper, and proven by cardboard models. These models will at once show any error in the patterns which might otherwise be overlooked. As only the Examination Plates are to be sent to the School, the student should draw all the other plates given in this course.

The first problem to be drawn is shown in Fig. 2, being the intersection between a cylinder and octagonal prism. In drawing these problems for practice, make the cylinder and octagonal prism both 2 inches in diameter. The height of the cylinder from B to E should be 4½ inches; and the length of the prism from G to H, 3 inches. Let A represent the plan of the cylinder, shown in elevation by B C D E; and F, the section of the prism, shown in plan by G H I J. Number the corners of the section F as shown, from 1 to 4 on both sides; and from these points draw horizontal lines intersecting the plan of the cylinder at 2′3′ and 1′4′ on both sides as shown. Establish a convenient intermediate point of intersection between the corners of the prism, as *a* and *a* in A, from which draw horizontal lines intersecting the section F at a', a', a', and a'. Take a tracing of the section F with its various intersections, and place it in its proper position as shown by F^1, in the

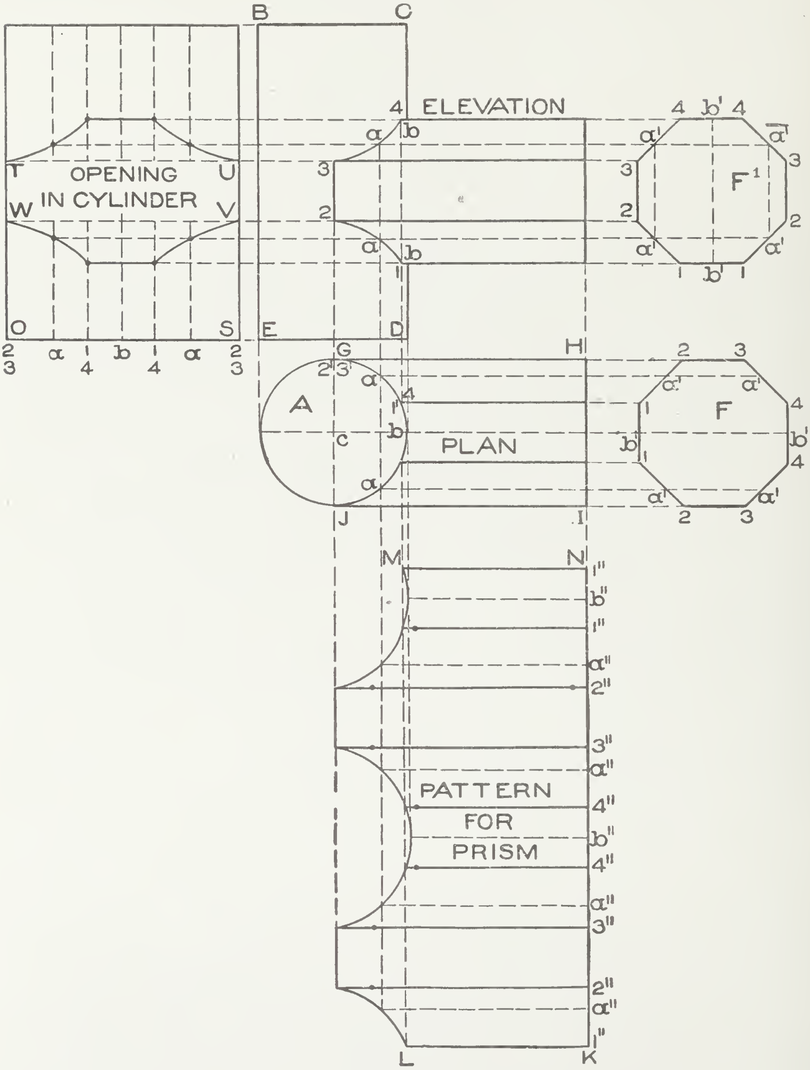


Fig. 2.

center of the cylinder B C D E, allowing the section to make a quarter turn, and bringing the points b' b' at the top and bottom on a vertical line, while in the section F, b' b' are on a horizontal line. From the various intersections in F^1, draw horizontal lines intersecting vertical lines drawn from similarly numbered intersections in the plan A, as shown in elevation. A line drawn through these points will represent the joint between the cylinder and prism.

For the development for the prism, extend the line H I in plan as N K, upon which place the stretchout of all the points contained in the section F, as shown by similar figures and letters on N K. Through these points, at right angles to N K, draw lines which intersect with lines drawn from similarly numbered points and letters in plan, at right angles to J I. Trace a line through points thus obtained, and K L M N will be the desired pattern. To obtain the development for the opening in the cylinder, extend the line D E in elevation as S O, upon which place the stretchout of all the points contained in the half-circle A, as shown by similar numbers and letters on S O. At right angles to S O and through these points, draw lines intersecting horizontal lines drawn from intersections having similar numbers and letters in elevation, thus obtaining the intersections shown by T U V W, which will be the shape of the opening to be cut into one-half of the cylinder.

In Fig. 3 is shown the intersection between a hexagonal and quadrangular prism, the hexagonal prism being placed in elevation at an angle of 45° to the base line. When drawing this problem for practice, make the height of the quadrangular prism 4½ inches, and each of its sides 2 inches. Place the hexagonal prism at an angle of 45° to the base line, placing it in the center of the quadrangular prism in elevation as shown; and inscribe the hexagonal section in a circle whose diameter is 2½ inches. Let A represent the plan of the quadrangular prism placed diagonally as shown, above which draw the elevation B C D E. In its proper position and proper angle, draw the outline of the hexagonal prism as shown by 1^{v} 1″ 4″ 4^{v}; and on 1″ 4″ draw the half section as shown by F, numbering the corners 1″ 2″ 3″ and 4″. From the corner 1′ in the plan A, draw the center line 1′ 4. Take a tracing of the half section F, and place it as shown by F^1, placing the points 1″ 4″ in F on the center line in F^1 as shown. From the corners 1, 2, 3, and 4, draw lines parallel to the center line, intersecting the two sides of A (*b* 1′ and 1′ *a*) at 2′ 3′ and 1′ 4′, as shown. From these intersections draw vertical lines, which intersect by lines drawn parallel to $4'' \; 4^v$ from corners having similar numbers in F, thus obtaining the points of intersection 1^v 2^v 3^v and 4^v. Dropping vertical lines from the intersections on the plane $1'' \; 4''$ in elevation, and intersecting similarly numbered lines in plan, will give the horizontal section of $1'' \; 4''$, as shown by 1° 2° 3° and 4°.

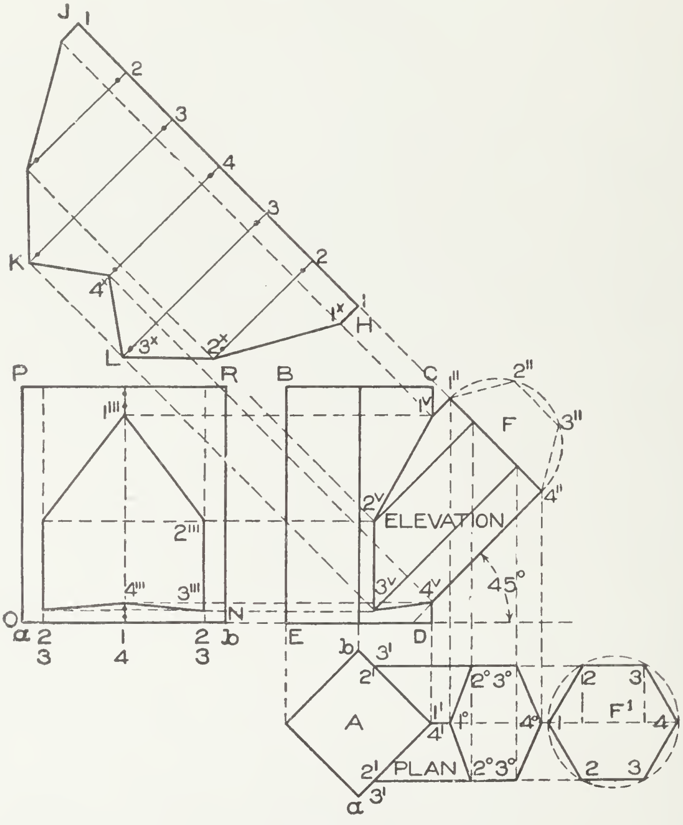


Fig. 3.

For the development of the hexagonal prism, extend the line $4'' \; 1''$ as shown by H J, upon which place the stretchout of twice the number of spaces contained in the half section F, as shown by similar figures on the stretchout line H J. From these points, at right angles to H J, draw lines as shown, which intersect by lines drawn at right angles to the line of the prism from intersections 1^v to 4^v, thus obtaining the points of intersection 1^x to 4^x. Lines traced from point to point as shown by J K L H, will be the required development. The shape of the opening to be cut into the quadrangular prism, is obtained by extending the line D E in elevation as N O, upon which place the stretchout of one-half the section A, with the various points of intersection, as shown by similar figures on O N. At right angles to O N erect lines from these points, which intersect by lines drawn from similarly numbered intersections in elevation at right angles to the quadrangular prism, thus obtaining the points of intersection $1'''$ to $4'''$ on both sides. Then N O P R will be the half development.

Fig. 4 shows the intersection between two cylinders of equal diameters at right angles. Make the height of the vertical cylinder 3 inches, that of the horizontal cylinder $1\frac{1}{4}$ inches, and the diameters of both 2 inches. Let A represent the plan of the vertical cylinder, and B its elevation. Draw the plan of the horizontal cylinder C, shown in elevation by D placed in the center of the vertical cylinder. Draw the half section E in plan and divide it into equal parts, as shown from 1 to 3 to 1. In a similar manner draw the half section E^1 in elevation, which also divide into the same number of spaces as E, reversing the numbers as shown.

The following suggestions are given to avoid confusion in numbering the points or corners of irregular or round sections in plan and elevation. If the half section E were bent on the line 1–1 and turned upward toward the reader, and we should view this section from the front, the point 3 would be at the top, or, if bent downward, would be at the bottom; therefore the points 3 and 3 in elevation are placed at top and bottom. Now if the section E^1 in elevation were bent on the line 3-3 either toward or away from the reader, the point 1 when looking down would show on both sides as shown in plan, which proves both operations. No matter whether the form is simple, as here shown, or complicated as that which will follow, the student should use his imaginative power. Study the problem well; close your eyes and imagine you see the finished article before you, or, failing in this, make a rough model in the shop or a cardboard model at home, which will be of service. Now from the intersections in E, draw horizontal lines intersecting the circle A at $1'$, $2'$ and $3'$ on both sides. From these points erect perpendicular lines and intersect them with horizontal lines drawn

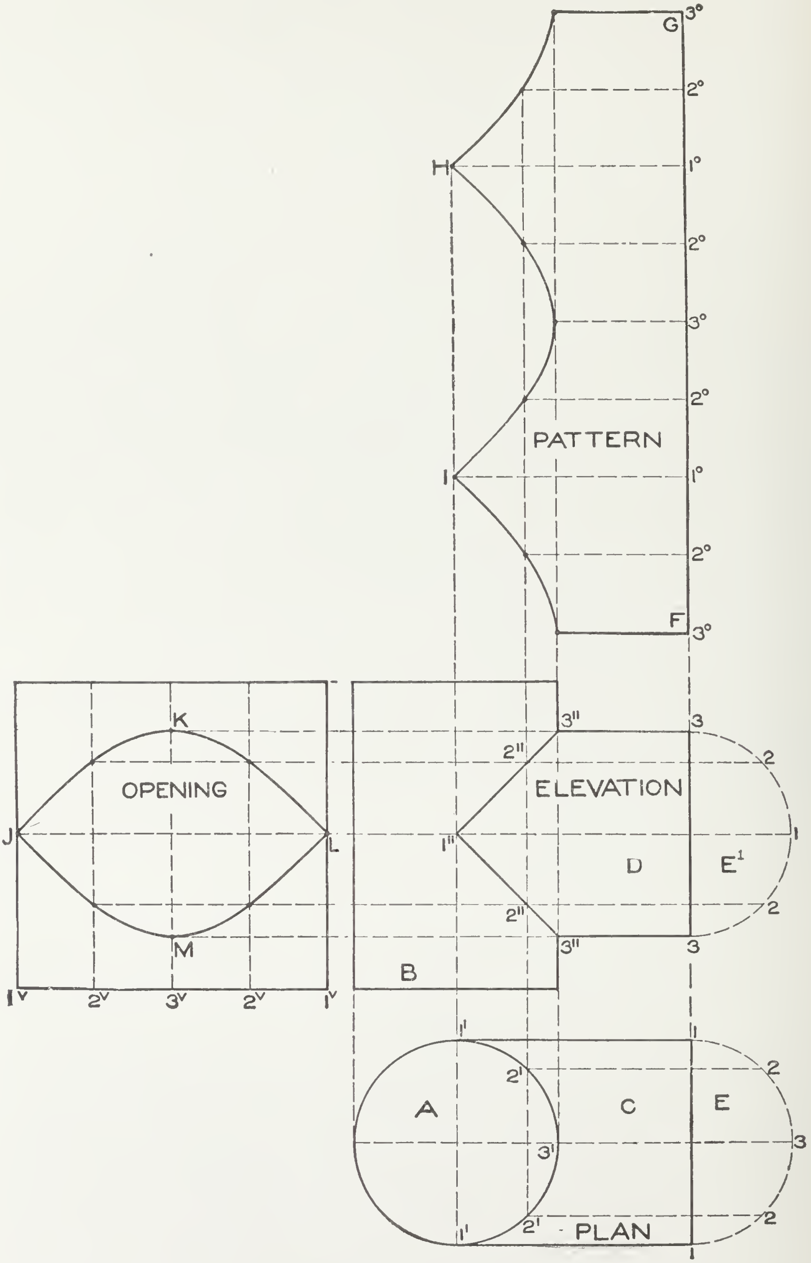

3°
G
2°
H
1°
2°
3°
2°
PATTERN
I
1°
2°
F
3°
K
3''
3
2''
2
OPENING
ELEVATION
J
L
1''
1
D
E1
2''
2
M
3''
3
B
1v
2v
3v
2v
1v
1'
1
2'
2
A
C
E
3'
3
1'
2'
PLAN
2

Fig. 4.

from similarly numbered intersections in E^1. Lines traced through these points 3″ 2″ 1″ and 1″ 2″ 3″ will be straight because both branches are of equal diameters.

For the development of the cylinder D in elevation, extend the line 3-3 as shown by F G, upon which place the stretchout of twice the number of spaces contained in E^1, as shown by similar numbers 3° to 1° to 3° to 1° to 3° on the stretchout line F G. From these points, at right angles to G F, draw lines, and intersect them by lines drawn parallel to the cylinder B from similar numbers in the joint line. Trace a line through these points in the development, when F G H I will be the desired shape.

For the opening to be cut into the cylinder B to receive the cylinder D, extend the base of the cylinder B as shown by 1^v 1^v, upon which place the stretchout of the half circle A in plan, as shown by similar figures on the stretchout line 1^v 1^v. From these points erect perpendiculars, which intersect by lines drawn from similarly numbered intersections in elevation at right angles to the line of the cylinder B. Trace a line through the intersections thus obtained; J K L M will be the shape of the opening.

Fig. 5 shows the intersection of two cylinders of unequal diameters at an angle of 45°. Make the diameters of the large and small cylinders 2 inches and $1\frac{1}{4}$ inches respectively; the height of the large cylinder 3 inches; and the length of the small cylinder measured from its shortest side in elevation, 1 inch, placed at an angle of 45° in the center of the cylinder B. A represents the plan of the large cylinder struck from the center a and shown in elevation by B. Draw the outline of the small cylinder C at its proper angle, and place the half section D in its position as shown; divide it into a number of equal spaces, as shown from points 1 to 5. Through the center a in plan. draw the horizontal line a 5; and with b as a center describe a duplicate of the half section D with the various points of intersection, as shown by D^1, placing the points 1 and 5 on the horizontal line a 5. From the intersections in D^1 draw horizontal lines intersecting the large circle A at 3′ to 3′ as shown, from which points erect perpendicular lines; intersect them by lines drawn parallel to the lines of the smaller pipe from similarly numbered intersections in D. A line traced through the points thus obtained will represent the intersection or miter joint between the two pipes.

These same principles are applicable no matter what diameters the pipes have, or at what angle they are joined, or whether the

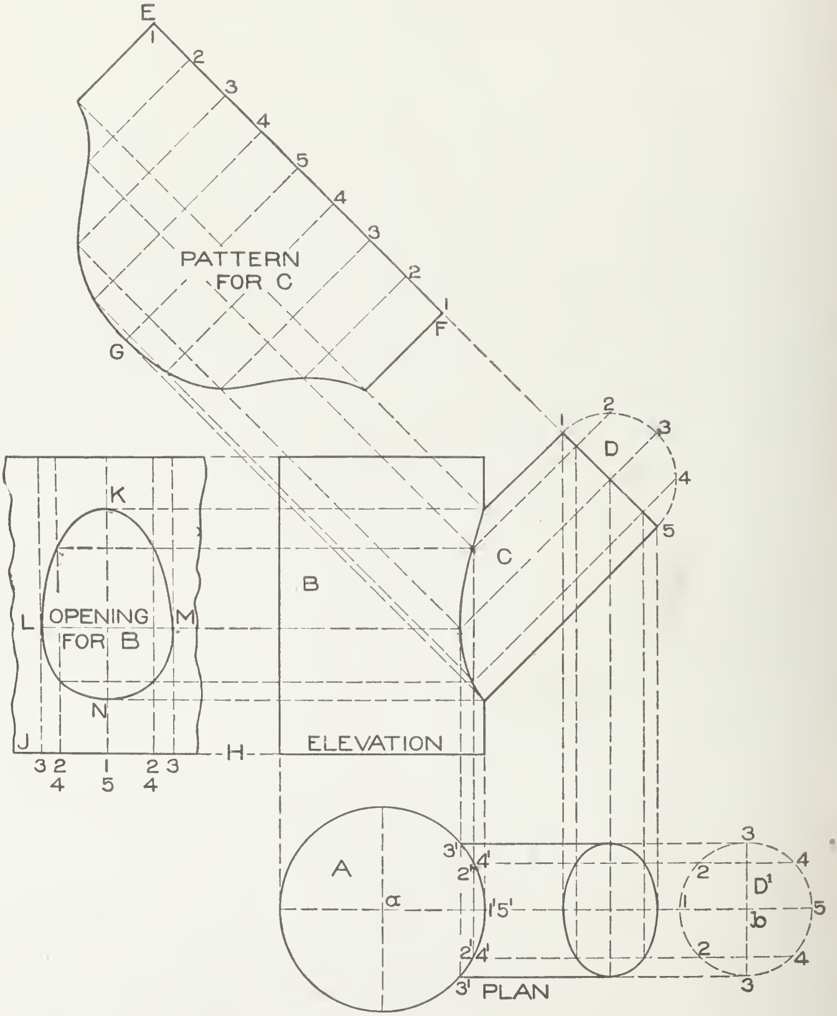


Fig. 5.

pipe is placed as shown in plan or at one side of the center line.

For the development of the small cylinder extend the line 5 – 1 in elevation as shown by F E, upon which place the stretchout of the circle D^1 in plan, or twice the amount of D in elevation, as shown by similar figures on the stretchout line F E. At right angles to F E and through these small figures, draw lines which intersect with lines drawn at right angles to the lines of the small cylinder from similarly numbered intersections in the miter line in elevation. Trace a line through the points thus obtained; E F G will be the development for the cylinder C.

To obtain the opening in the large cylinder extend the lines of the large cylinder in elevation as shown at the base by H J, upon which place the stretchout of the intersections contained in the circle A, being careful to transfer each space separately (as they are unequal) to the stretchout line H J. Through these points and at right angles to H J erect lines which intersect with horizontal lines drawn from similar points in the miter line in elevation A line traced through the points thus obtained, as shown by K L M N, will be the desired development.

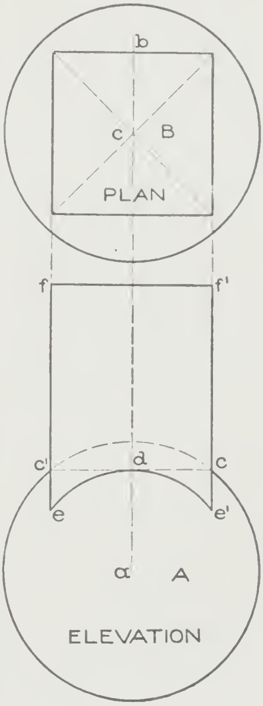


Fig. 6.

Fig. 6 shows the intersection between a quadrangular prism and sphere, the center of the prism to come directly over the center of the sphere. Make the diameter of the sphere $2\frac{1}{2}$ inches, the sides of the prism $1\frac{1}{2}$ inches, and the height from f to c' $2\frac{3}{8}$ inches. Draw the elevation of the sphere A which is struck from the center a, from which erect the perpendicular $a\ b$. With any point, as c, as a center and using the same radius as that used for A, describe the plan B. Through c draw the two diagonals at an angle of 45°, and draw the plan of the prism according to the measurements given. Now draw the elevation of the prism $f\ c'$ and $f''\ c$, the sides of the prism intersecting the sphere at c and c'. From either of these points draw a horizontal line intersecting the center line $a\ b$ at d. Then using a as a center and $a\ d$ as the radius, describe the arc $e\ e'$ intersecting the sides of the prism extended at e and e'; $f\ e\ e'\ f''$

will be the development for one of the sides of the prism. In practice the four sides are joined in one.

Fig. 7 shows the intersection of a quadrangular prism and sphere when the center of the prism is placed to one side of the center of the sphere. Make the diameter of the sphere the same as in the preceding figure; through *x* in the plan draw the 45° diagonal, and make the distance from *x* to A $\frac{1}{2}$ inch, the sides of the prism 1 inch, and the height from E to *c* in elevation $1\frac{1}{2}$ inches. Having drawn the elevation and plan of the sphere, construct the plan of the prism as shown by A B C D. Parallel to the center line *x y* project the prism in elevation intersecting the sphere at *a* and *c*. Now since the center of the sphere is on one of the diagonals of the prism in plan, either two of the sides meeting at one end of that diagonal, as B C and C D, will be alike, and both will be different from the other two sides A B and A D, meeting at the opposite end of the diagonal. Therefore the line F *a* in elevation will be used in obtaining the development of D C in plan, while the line E *c* will be used in obtaining the development for the two sides D A and A B in plan.

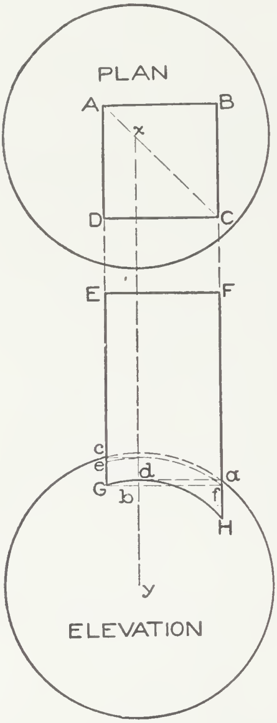


Fig. 7.

Now from *a* draw a horizontal line intersecting the center line *x y* at *b;* and using *y* as a center and *y b* as the radius, describe the arc G H intersecting the sides of the prism extended to G and H. Then E F G H is the development for each side of the prism shown in plan by D C and C B. In a similar manner, from the intersection *c* in elevation draw a horizontal line intersecting the center line *x y* at *d*. Then using *y* as center and *y d* as radius, describe an arc intersecting the sides of the prism at *e* and *f*. E F *f e* will show the development for either side of the prism shown in plan by D A and A B. By connecting the points G and *f* it will be found that the line is a true horizontal line, which proves the two developments. Should the plan of the prism be so placed on the sphere that all sides would be different, then two elevations would be necessary so that the intersections of all the sides could be shown.

Developments by Triangulation. In developing sheet-metal work of irregular forms, patterns are required which cannot be developed by either the parallel or radial-line methods. These irregular shapes are so formed that although straight lines can be drawn upon them the lines would not run parallel to one another, nor would they all incline to a common center. In the methods previously described, the lines in parallel developments run parallel to one another, while in radial-line developments all the lines meet at a common center. Hence in the development of any irregular article, it becomes necessary to drop all previous methods, and simply proceed to measure up the surface of the irregular form, part by part, and then add one to another until the entire surface is developed. To accomplish this, one of the simplest of all geometrical problems is made use of and shown in Part II of Mechanical Drawing, Plate V, Problem 11, entitled "To construct a triangle having given the three sides." To carry out this method it is necessary only to divide the surface of the plan or elevation of any irregular article into a number of equal parts. Use the distances in plan as the bases of the triangles, and the distances in elevation as the altitudes or heights of the triangles, or *vice versa;* and then find the hypothenuse by connecting the two given lengths.

To illustrate this simple principle Fig. 8 has been prepared. Let A B C D represent the plan of a plane surface, shown in elevation by A^1 B^1. We know that the true length of the plane is equal to A^1 B^1 and the true width is equal to A D or B C in plan. We also know that the vertical height from the bottom of the plane A^1 to the top B^1 is equal to B^1 *b* as shown. But suppose we want to obtain the true length of the diagonal line B D in plan on the developed plane. To obtain this it will be necessary only to take the length of B D, place it from *b* to D^1, and draw a line as shown from B^1 to D^1, which is the length desired.

While this may look very simple, it is all that there is to triangulation, and if the student thoroughly understands the simple principle and studies the problems which will follow, he will have no trouble in applying this principle in complicated work. To make it still clearer we will prove the length of the line B¹ D¹. Take the distance of A¹ B¹, place it in plan as shown by A B², and complete the rectangle A B² C² D. Draw the diagonal B² D, being the length sought, which will be found to equal B¹ D¹ in elevation. When drawing this problem in practice, make the plan 4 by 6 inches and the vertical height in elevation 5 inches.

In obtaining developments by triangulation, the student should use all of his conceptive powers as previously explained. Before making any drawing, he must see the article before him in his mind's eye, so to speak, before he can put it down on paper. Therefore we want to impress upon the student the necessity of drawing all the problems that will follow in this part and in the Practical Workshop Problems. It should be understood that triangulation is not given as an alternative method, but is used when no other method can be employed, and without it no true pattern could be obtained for these irregular shapes; hence the necessity of close study.

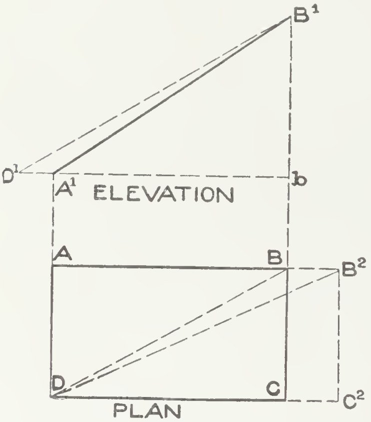


Fig. 8.

In Fig. 9 is shown an irregular solid whose base and top are triangles crossing each other, and in which the principle just explained will be put to practical test Inscribe the triangles shown in plan in a circle whose radius is equal to a 1, or 1½ inches, and make the height of the article in elevation 2 inches. The dotted triangles 1 2 3 in plan represent the section of the article on the line 2–3 in elevation: and the solid triangle 1^1 2^1 3^1 in plan, the section on the line 2^1 3^1 in elevation. Now connect the two sections in plan by drawing lines from 1 to 2^1 and to 3^1, from 2 to 2^1 and to 1^1, and from 3 to 1^1 and to 3^1. In a similar manner connect the points in elevation as shown. It now becomes necessary to obtain a triangle giving the true length of the lines connecting the corners of the triangle in plan, and as all of these lines are equal only one triangle is necessary. Therefore take the distance from

1 to 2^1 in plan and place it on the line 3–2 extended in elevation, as shown from 2 to 1°, and draw a line from 1° to 2^1, which is the desired length.

For the pattern, proceed as is shown in Fig. 10. Take the distance of any one of the sides in the triangle, as 1–2 in Fig. 9, and place it on the horizontal line 1–2 in Fig. 10. Then using 1 and 2 as centers, with 1° 2^1 in elevation in Fig. 9 as radius, describe the arcs in Fig. 10 intersecting each other in 2^1. Then 1 2 2^1 will be the pattern for one of the sides shown in plan in Fig. 9 by 1 2 2^1. Proceed in this manner in Fig. 10 as shown by the small arcs; or a tracing may be taken of the one side 1 2 2^1, and traced as shown until six sides are obtained, which will be the full pattern and which is numbered to correspond to the numbers in plan.

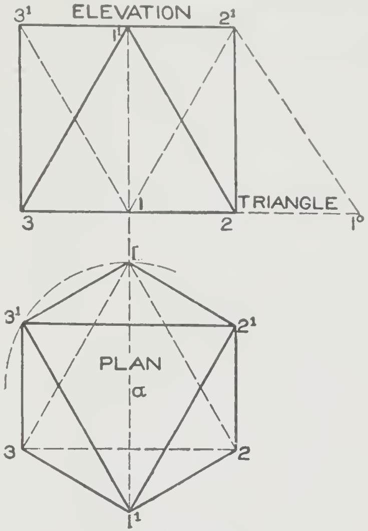


Fig. 9.

In Figs. 11, 12, and 13 are shown the methods used in developing a scalene cone. The method of obtaining the development of any scalene cone, even though its base is a perfect circle, is governed by the same principle as employed in the last problem on triangulation It is well to remember that any section of a scalene cone drawn parallel to its base will have the same shape (differing of course in size) as the base. This is equally true of articles whose

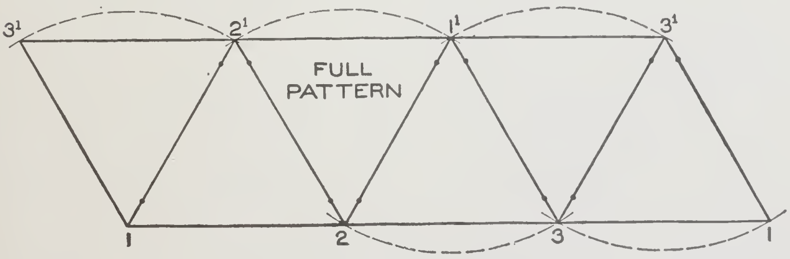


Fig. 10.

bases are in the shape of a square, rectangle, hexagon, octagon, or any other polygon. What has just been explained will be proven in connection with Fig. 11, in which A B C represents a side elevation of a scalene cone, whose plan is shown by 1 4^1 7 4 C^1. Draw any horizontal line, as A D, on which set off the distances A B equal to 3 inches and B D equal to $2\frac{1}{2}$ inches, and the vertical height D C equal to $4\frac{1}{2}$ inches. Draw lines from B and A to C, which completes the elevation. In its proper position below the line A B, draw the plan of A B as 1 4 7 4^1 struck from the center C. Through C draw the horizontal line C C^1, and

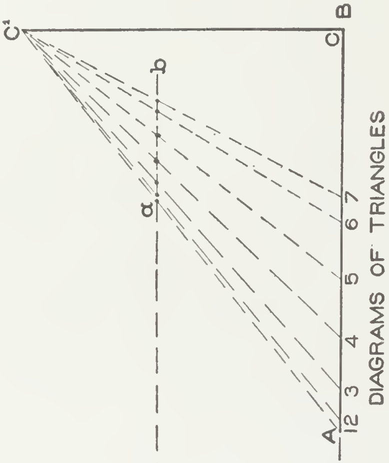


Fig. 12.

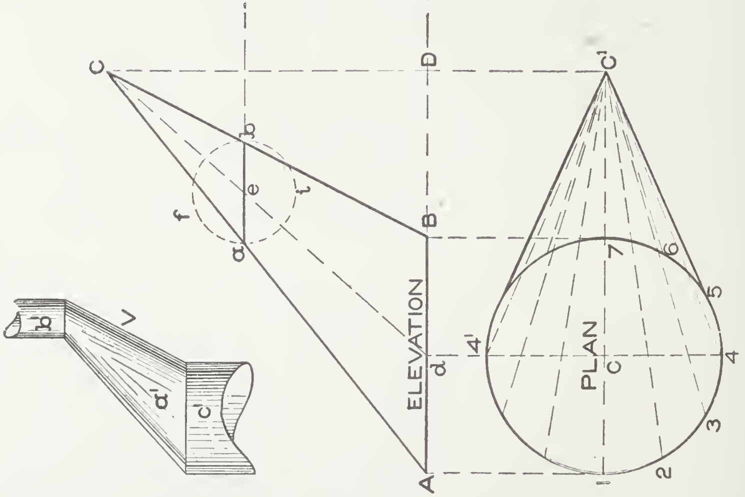


Fig. 11.

intersect it by a vertical line drawn from the apex C in elevation, thus obtaining the apex C^1 in plan. Draw lines from 4 and 4^1 to C^1, which completes the plan.

As both halves of the scalene cone are symmetrical, it is necessary only to divide the half plan 1 4 7 into a number of equal spaces as shown by the small figures 1 to 7, and from points thus obtained draw radial lines to the apex C^1. Then these lines in plan will represent the bases of triangles which will be constructed, whose altitudes are all equal to D C in elevation. Therefore in Fig. 12 draw any horizontal line, as A B, and from any point, as C, erect the perpendicular line C C^1 equal in height to D C in Fig. 11. Now from C^1 in plan take the various lengths of the lines 1 to 7 and place them on the line A B in Fig. 12, measuring in every instance from the point C, thus obtaining the intersections 1 to 7, from which lines are drawn to the apex C^1. Then these lines will represent the true lengths of similarly numbered lines in plan in Fig. 11.

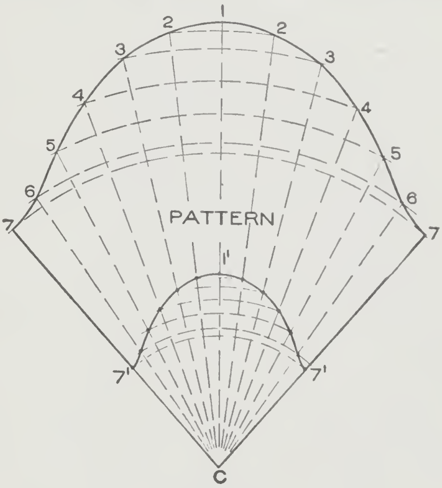


Fig. 13.

For the pattern proceed as is shown in Fig. 13. With C as center and radii equal to C^1 7, 6, 5, 4, etc., in Fig. 12, describe the arcs 7–7, 6–6, 5–5, 4–4, etc., in Fig. 13 as shown. Now assuming that the seam is to come on the short side of the cone, as C B in Fig. 11, set the dividers equal to one of the equal spaces in the plan; and starting on the arc 7–7 in Fig. 13, step from arc 7 to arc 6, to arcs 5, 4, 3, 2, and 1, and then continue to arcs 2, 3, etc., up to 7. Trace a line through these intersections as shown by 7–1–7, and draw lines from 7 and 7 to C, which completes the pattern.

Now to prove that any section of an oblique or scalene cone cut parallel to its base, has a similar shape to its base (differing in size), draw any line as *a b* in Fig. 11 parallel to A B. From C in plan erect a vertical line intersecting the base line A B at d, from which draw a line to the apex C, cutting the line $a\ b$ at e. Then the distances $e\ a$ and $e\ b$ will be equal; and using e as a center and $e\ b$ as radius, describe the circle $a\ f\ b\ i$, which is the true section on $a\ b$. Then $a\ b$ B A will be the frustum of a scalene cone. Extend the line $a\ b$ parallel to A D, cutting the diagram of triangles in Fig. 12 from a to b. Then with radii equal to the distances from C^1 to the various intersections on the line $a\ b$, and using C in Fig. 13 as center, intersect similarly numbered radial lines drawn from 7 to 1 to 7 to the apex C. A line traced as shown from $7'$ to $1'$ to $7'$ will be the desired cut, and $7–7–7'–7'$ will be the pattern for the frustum. The practical use of this method is shown in diagram V in Fig. 11; a' is the frustum of the oblique cone, on the ends of which are connected round pipes b' and c'.

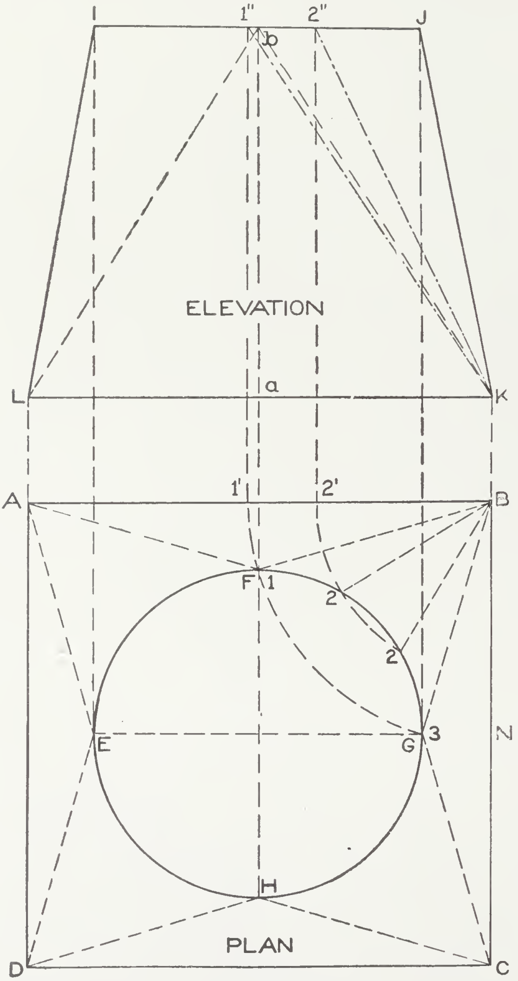


Fig. 14.

It is shown in Fig. 14 how in an irregular solid whose base is square and top is round, both top and bottom on horizontal planes are developed. The corners in plan F B G, G C H, H D E and E A F should be considered as sections of scalene cones. Proceed by drawing the plan A B C D $3\frac{1}{2}$ inches square, which represents the plan of the base of the article; and the circle E F G H 2½ inches in diameter, which shows the plan of the top of the article; the vertical height to be 3 inches, shown from *a* to *b*. As the circle is in the center of the square, making the four corners symmetrical, it is necessary only to divide the one-quarter circle into a number of equal parts as shown by the small figures 1, 2, 2, 3, from which draw lines to the apex B. Complete the elevation as shown by I J K L. Now using B as center, and radii equal to B 1 and B 2 in plan, describe arcs intersecting A B at 1′ and 2′ as shown. From these points erect perpendiculars intersecting the top of the article I J

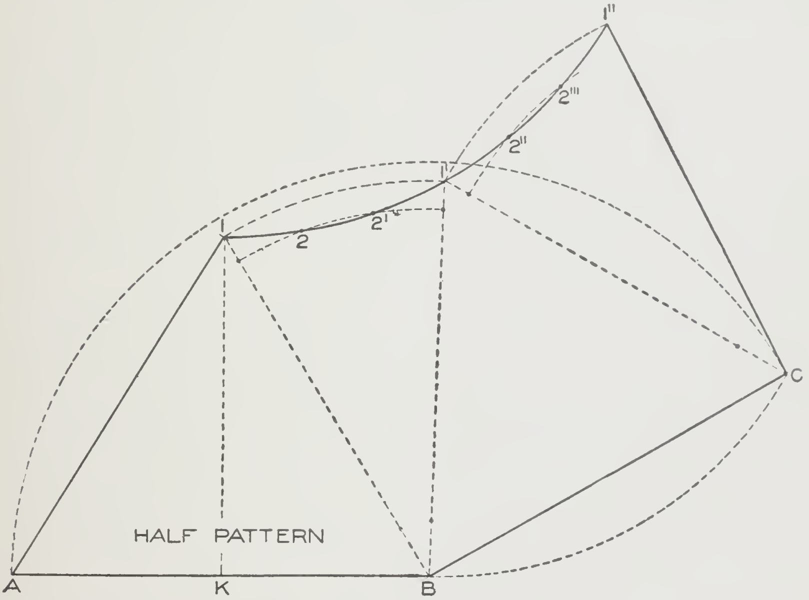


Fig. 15.

in elevation at 1″ and 2″, from which draw lines to K. Then K 1″ and K 2″ will be the true lengths of the lines shown in plan by B 1 and B 2 respectively on the finished article.

For the half pattern proceed as follows: In Fig. 15 draw any horizontal line, as A B, equal in length to A B in plan in Fig. 14. Now with K 1″ as radius and A and B in Fig. 15 as centers, describe arcs intersecting each other at 1 From 1 drop a vertical line intersecting A B at K. Then 1 K should equal J K in elevation in Fig. 14, which represents the true length through G N in plan.

Now with radii equal to K 1″ and K 2″ in elevation, and with B in Fig. 15 as center, describe the arcs 1–1′ and 2–2′. Now set the dividers equal to one of the spaces in G F in plan in Fig. 14; and starting at 1 in Fig. 15, step off arcs having similar numbers as shown by 1, 2, 2′, 1′. Now using 1 B as radius, and 1′ as center, describe the arc B C, and intersect it by an arc struck from B as center and with B A as radius, as shown at C. Take a tracing of 1 B 1′ and place it as shown by 1′ C 1″. Now connect the various intersections by drawing lines from 1 to A to B to C to 1″ to 1′ to 1, which completes the half pattern. The triangular pieces 1 A B or 1′ B C will represent the flat sides of the article shown in plan by 1 A B or 3 B C respectively in Fig. 14; and the cone patterns 1–1′ B and 1′–1″ C in Fig. 15, the sections of the scalene cones 1–3–B and H–G–C respectively in plan in Fig. 14. This same rule is applicable whether the top opening of the article is placed exactly in the center of the base or at one side or corner. Various problems of this nature will arise in Practical Workshop Problems; and if the principles of this last problem are thoroughly understood, these will be easily mastered.

Approximate Developments. In developing the blanks or patterns for sheet-metal work which requires that the metal be hammered or raised by hand, or passed between male and female dies in foot or power presses, circular rolls, or hammering machines, the blanks or patterns are developed by the approximate method, because no accurate pattern can be obtained. In all raised or pressed work in sheet metal, more depends upon the skill that the workman has with the hammer, than on the patterns, which are but approximate at their best. While this is true, it is equally true that if the workman understands the scientific rule for obtaining these approximate patterns a vast amount of time and labor can be saved in bringing the metal to its proper profile. If the true rule for averaging the various shapes and profiles in circular work is not understood, the result is that the blank has either too little or too great a flare and will not form to its proper profile and curve. Before proceeding to describe the approximate development methods, attention is called to the governing principle underlying all such operations. We have previously shown how the patterns are developed for simple flaring ware; in other words, how to develop the frustum of a cone. The patterns for curved or any other form of circular or hammered work are produced upon the same principle. The first illustration of that principle is shown in Fig. 16, in which A B C D represents a sphere 3 inches in diameter composed of six horizontal sections, struck from the center *a*.

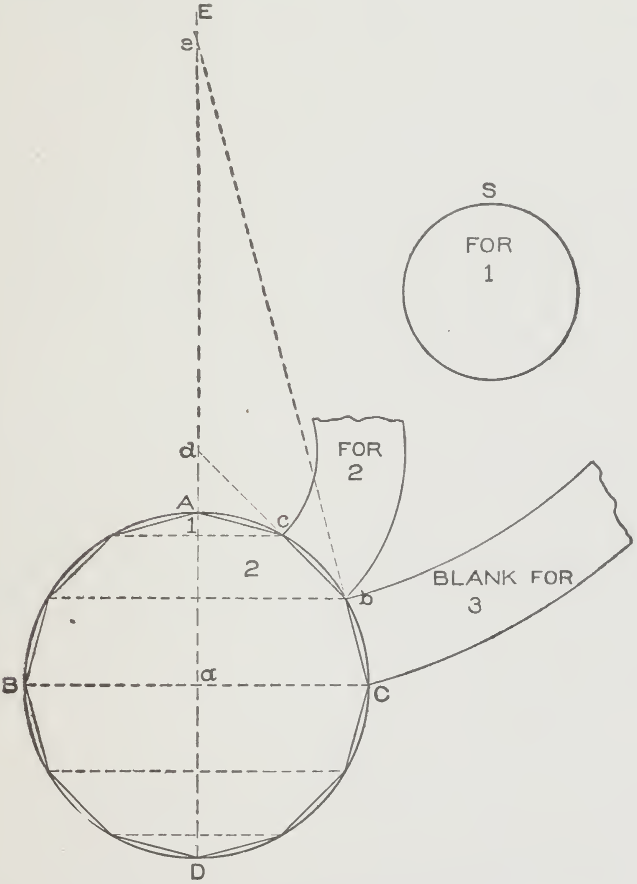


Fig. 16.

Divide the quarter circle A C into as many parts as there are sections required in the half sphere (in this case three), and draw horizontal lines through the ball as shown. The various radii for the patterns are then obtained by drawing lines through C *b*, *b c*, and *c* A. Thus C *b* extended meets the center line E D at *e*, which is the center for striking the blank for number 3, using the radii *e b* and *e* C. In similar manner draw a line from *b* to *c*, extending it until it meets E D at *d*. Then *d c* and *d b* will be the radii for blank number 2, while A *c* is the radius for blank 1 shown at S The lengths of the pattern pieces are determined in the same manner as would be the case with an ordinary flaring pan in producing the patterns for tin ware, and will be explained

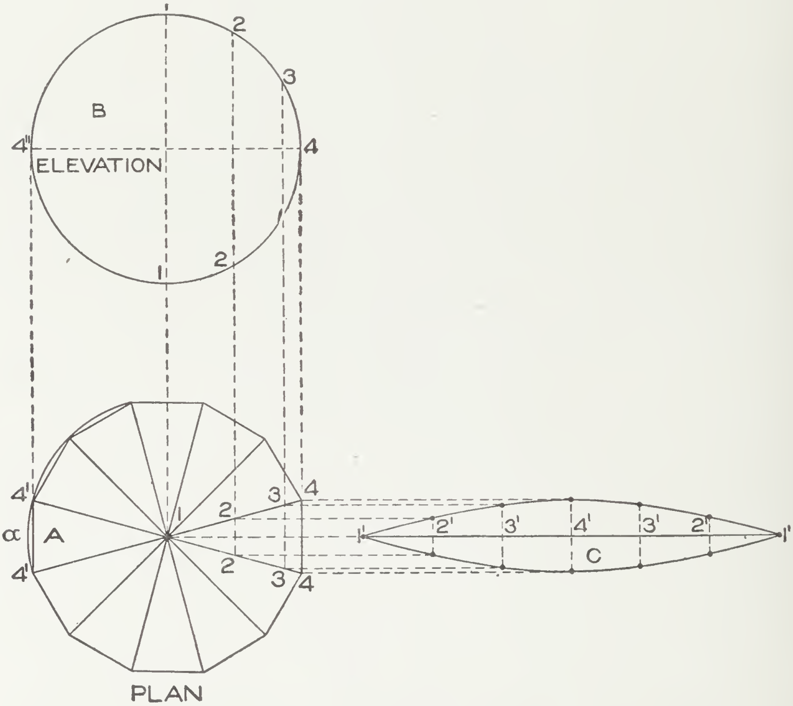


Fig. 17.

thoroughly in the Practical Workshop Problems which will shortly follow.

In Fig. 17 is shown another elevation of a sphere composed of twelve vertical sections as shown in plan view. While the method used for obtaining the pattern is by means of parallel lines, and would be strictly accurate if the sections in plan remained straight as from 4 to 4, the pattern becomes approximate as soon as we start to raise it by means of machine or hammer to conform to the profile B in elevation, because the distance along the curve *a* from 4′ to 4′

in plan is greater than a straight distance from 4 to 4. The pattern by this method is obtained as follows: Let B represent the elevation of the sphere, and A the plan of the same, which is divided into as many sides as the sphere is to have vertical sections, in this case 12, being careful that the two opposite sides 4–4 and 4′ 4′ in plan run parallel to the center line as shown. Make the diameter of the sphere 4–4″ 3 inches. Divide the half elevation into an equal number of spaces as shown from 1 to 4 to 1, and from these points drop lines at right angles to 4–4″ intersecting the miter lines 1–4 in plan as shown. Now draw any horizontal line, as 1′–1′, upon which place the stretchout of 1–4–1 in elevation as shown by 1′–4′–1′ on the line 1′–1′ in C. Through these points draw lines at right angles to 1′–1′, which intersect by lines drawn from similarly numbered intersections on the miter lines 1–4 in plan, at right angles to 4–4. A line traced through points thus obtained as shown by C will be the desired pattern.

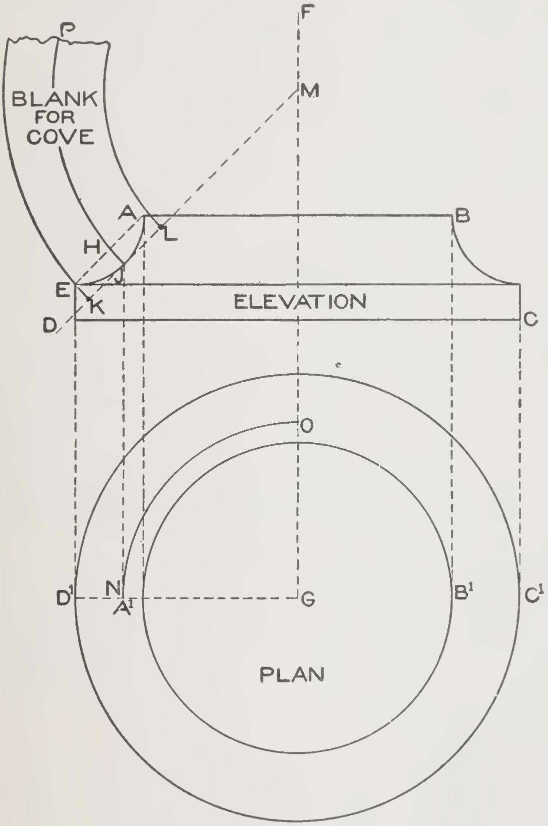


Fig. 18.

In Fig. 18 is shown the principle used in obtaining the radii with which to develop the blank for a curved or circular mould when it is to be hammered by hand. In this connection, only the principle employed will be shown, leaving the full development and also the development for patterns which are to be raised by hand

and hammered by machine, to be explained in problems which will follow in Practical Workshop Problems. Draw this problem double the size shown. First draw the elevation A B C D, and through the elevation draw the center line F G. Then using G as a center, draw the circles A^1 B^1 and C^1 D^1 representing respectively the horizontal projections of A B and C D in elevation. Now draw a line from A to E in elevation, connecting the corners of the cove as shown. Bisect A E and obtain the point H, from which at right angles to A E draw a line intersecting the cove at J. Through J parallel to A E draw a line intersecting the center line F G at M. Take the stretchout from J to A and from J to E and place it on the line J M as shown respectively from J to L and from J to K. Then will M L and M K be the radii with which to strike the pattern or blank for the cove. From J drop a vertical line intersecting the line D^1 G in plan at N. Then with G as center strike the quarter circle N O. Now using M as center and M J as radius, strike the arc J P. Then on this arc, starting from J, lay off 4 times the stretchout of N O in plan for the full pattern. It should be understood that when stretching the cove A E, the point J remains stationary and the metal from J to L and from J to K is hammered respectively toward J A and J E. For this reason is the stretchout obtained from the point J.

PRACTICAL WORKSHOP PROBLEMS.

In presenting the 32 problems which follow on sheet-metal work, practical problems have been selected such as would arise in every-day shop practice.

In this connection we wish to impress upon the student the necessity of working out each and every one of the 32 problems. Models should be made from stiff cardboard, or, if agreeable to the proprietor of the shop, the patterns can be developed at home, then cut out of scrap metal in the shop during lunch hour, and proven in this way.

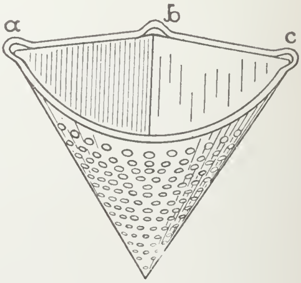


Fig. 19.

Our first problem is shown in Fig. 19, and is known as a sink drainer. It is often the case that the trap under the kitchen sink is choked or blocked, owing to a collection of refuse matter. To avoid this a sink drainer is used, and is fastened in position through the wire loops *a*, *b* and *c*. The refuse matter is poured into the drainer, from which it is easily removed after the fluid has passed through the perforations. These drainers may be made of tin or of black or galvanized iron, but where a good job is wanted 16-ounce copper should be used. To obtain the pattern for any sized drainer, proceed as follows: First draw the plan of the drainer A B C in Fig. 20, making A B and B C each two inches and forming a right angle. Then using B as center and A B as radius, draw the arc A C. In its proper position above the plan construct the side elevation, making E D 2 inches high, and draw the line F D. Then will F E D be the side elevation. Divide the arc A C into equal spaces as shown by the small figures 1 to 5. For the pattern use F D as radius, and with D in Fig. 21 as center strike the arc 1 5. From 1 draw a line to D and step off on 1–5 the same number of spaces as contained in A C in plan in Fig. 20, as shown by similar figures in Fig. 21. Draw a line from 5 to D. Then will 1–5–D be the pattern for the front of the strainer, in which perforations should be punched as shown. To join the sides of this pattern, use 1 and 5 as centers, and with either F E or A B in Fig. 20 as radius, describe the arcs E and E^1 in Fig. 21. Now using D as center and D E in Fig. 20 as radius, intersect the arcs E and E^1 as shown in Fig. 21. Draw lines from 1 to E^1 to D to E to 5, which completes the pattern, to which edges must be allowed for wiring at the top and seaming at the back.

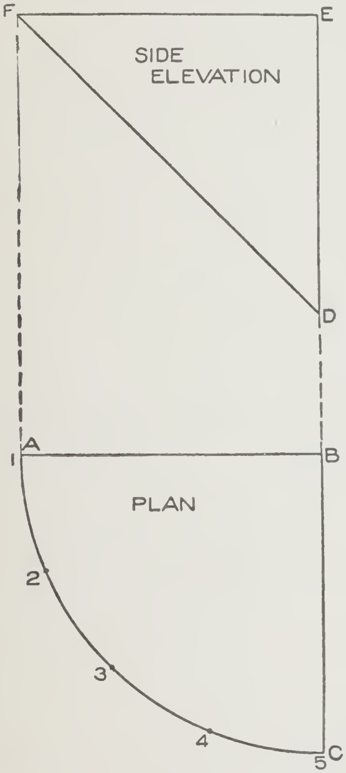


Fig. 20.

When joining a faucet or stop cock to a sheet-metal tank it is usual to strengthen the joint by means of a conical "boss," which is indicated by A in Fig. 22. In this problem the cone method is employed, using principles similar to those used in developing a frustum of a cone intersected by any line. Therefore in Fig. 23 let A B represent the part plan of the tank, C portion of the faucet extending back to the tank line, and F G H I the conical "boss" to fit around a faucet. When drawing this problem make the radius of the tank D A equal to $3\frac{1}{2}$ inches, and from D draw the vertical line D E. Make the distance from G to H equal to $2\frac{3}{4}$ inches, the diameter of the faucet F I $1\frac{1}{4}$ inches and the vertical height K C $1\frac{1}{2}$ inches Draw a line from G to H intersecting the center line D E at K. Then using K as center describe the half section G J H as shown. Divide J H into equal parts shown from 1 to 4, from which drop vertical lines intersecting the line G H as shown. from which draw radial lines to the apex E cutting the plan line

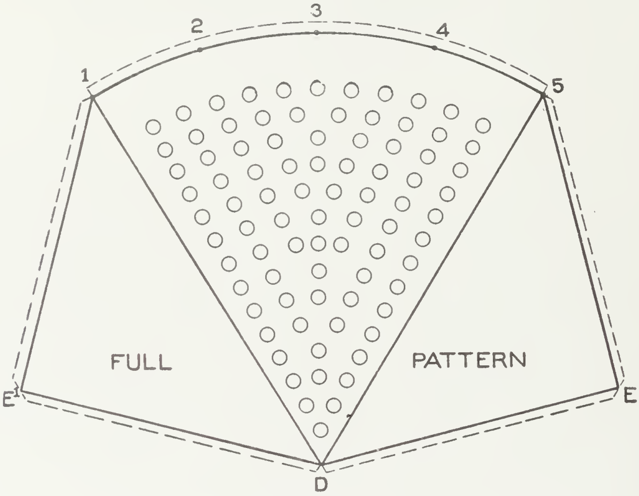


Fig. 21.

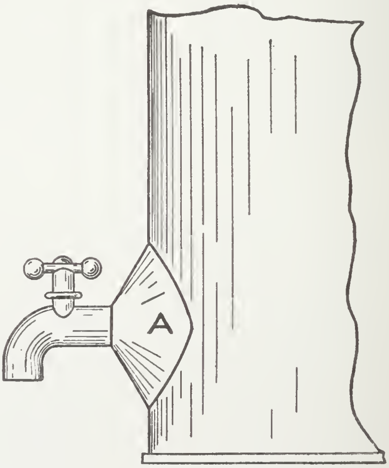


Fig. 22.

of the tank A B as shown. From these intersections draw horizontal lines intersecting the side of the cone H I at 1, 2′, 3′, and 4′. Now use E as center, and with radius equal to E 1 describe the

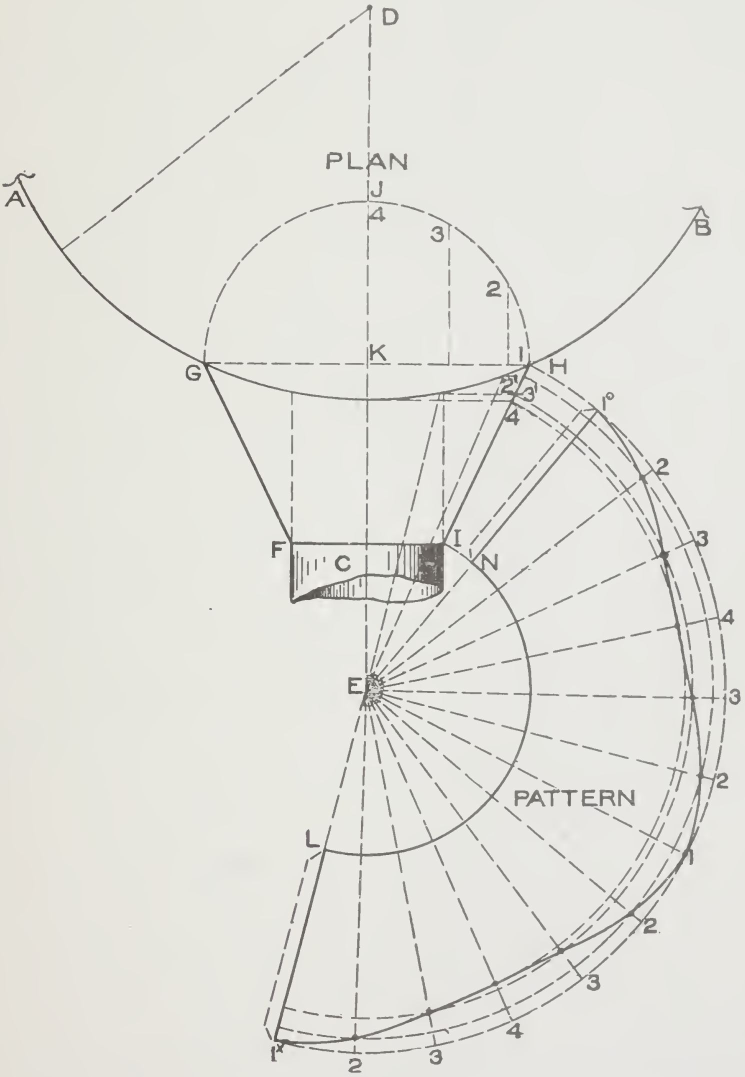


Fig. 23.

arc 1°–1^{x} as shown. Draw a line from 1^{x} to E, and starting from 1° set off on 1°–1^{x} four times the number of spaces contained in

J H in plan, as shown by similar numbers on 1° 1^{x}. Draw a line from 1^{x} to E, and with E I as radius describe the arc N L intersecting the radial lines 1° E and 1^{x} E at N and L respectively. From the various numbers on the arc 1° 1^{x} draw radial lines to the apex E; and using E as center and with radii equal to E 4′, E 3′, and E 2′, draw arcs intersecting similarly numbered radial lines as shown. Trace a line through points thus obtained; then will N 1° 1 1^{x} L be the pattern for the "boss."

In Fig. 24 is shown what is known as a hip bath. In drawing out the problem for practice the student should remember that it is similar to the preceding one, the only difference being in the outline of the cone. Make the top of the cone I B in Fig. 25 equal to $3\frac{1}{4}$ inches, the bottom C D $1\frac{3}{4}$ inches, the vertical height from K to 5′ $2\frac{1}{2}$ inches, the diameter of the foot E F $2\frac{1}{4}$ inches, and the vertical height 5′–5″ $\frac{1}{4}$-inch. Through the center of the cone draw the center line K L, and at pleasure draw the outline of the bath as shown by A J B. It is immaterial of what outline this may be, the principles that follow being applicable to any case. Thus, in the side elevation, extend the lines B C and A D until they intersect the center line at L. In similar manner extend the sides of the foot piece E D and F C until they intersect the center line at R. Now with 5′ as center and with radius equal to 5′ D or 5′ C, describe the half section C H D, which divide into equal spaces as shown by the small figures 1 to 9. From the points of division erect vertical lines meeting the base line of the bath D C at points 1, 2′, 3′, etc., to 9. From the apex L and through these points draw radial lines intersecting the outline B J A, from which horizontal lines are drawn intersecting the side of the bath B C as shown from 1 to 9. For the pattern for the body use L as center, and with L C as radius draw the arc F L^{1}. Now starting at any point, as 1, set off on F L^{1} twice the stretchout of D H C as shown by similar numbers on the arc F L^{1}. From the apex L and through the small figures draw radial lines, which intersect by arcs

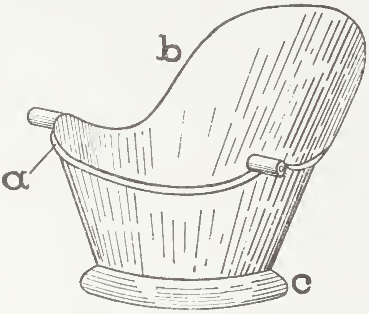


Fig. 24.

struck from L as center with radii equal to similarly numbered intersections on B C. Trace a line through points thus obtained, and L¹ M N P F will be the pattern for the body of the bath, to which laps should be added at the bottom and sides for seaming.

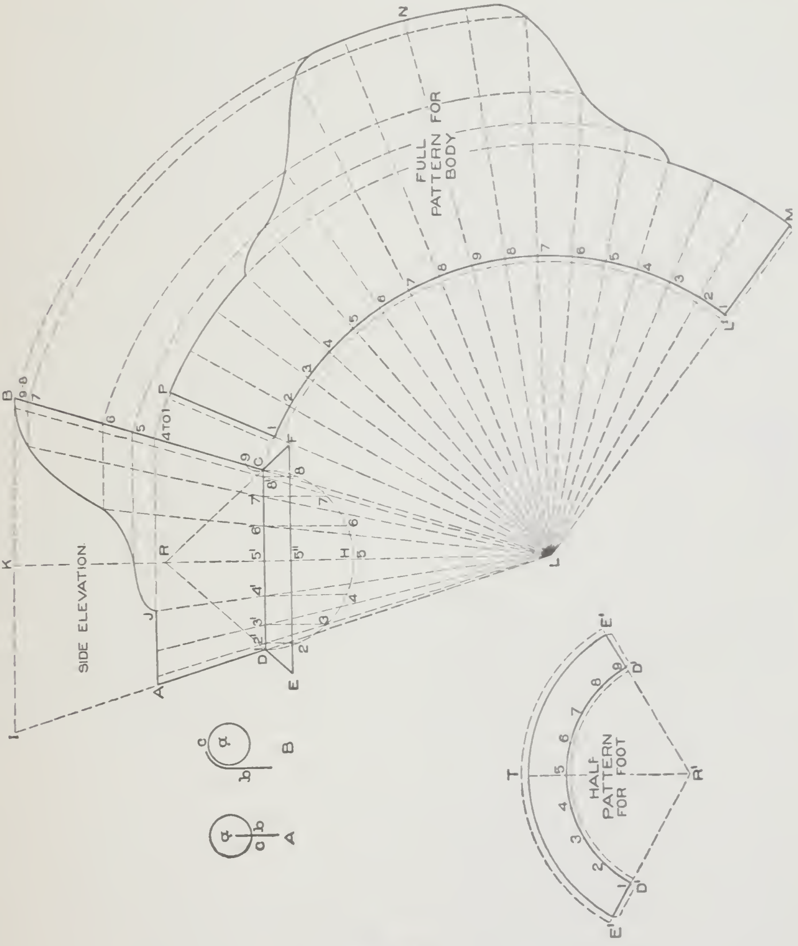


Fig. 25.

The pattern for the foot is obtained by using as radii R D and R E, and striking the pattern using R¹ as center, the half pattern being shown by E¹ T E¹ D¹ D¹, and the distance D¹ D¹ being equal to the stretchout of the half section D H C in side elevation.

It is usual to put a bead along the edges of the top of a bath as shown at *a* and *b* in Fig. 24. For this purpose tubing is sometimes used, made of brass, zinc, or copper and bent to the required shape; or zinc tubes may be rolled and soldered by hand, filled with heated white sand or hot rosin, and bent as needed. The tube or bead can be soldered to the body as shown in (A) in Fig. 25. Here *a* represents the bead, in which a slot is cut as *c*, and which is then slipped over the edge of the bath and soldered. Another method is shown in (B), in which the bath body *b* is flanged over the bead *a* and soldered clean and smooth at *c*, being then scraped and sandpapered to make a smooth joint. A wired edge is shown at *c* in Fig. 24, for which laps must be allowed as shown in Fig. 25 on the half pattern for foot.

In Fig. 26 is shown the perspective view of a bath tub; these tubs are usually made from IX tin or No. 24 galvanized iron. The bottom and side seams are locked and thoroughly soldered, while the top edge is wired with handles riveted in position as shown at A. The method used in developing these patterns will be the cone method and triangulation. In drawing this problem for practice (Fig. 27), first draw the center line W 8 in plan; and using *a* as center with a radius equal to $1\frac{1}{2}$ inches draw the semicircle C–12 D. Now make the distance *a* to *b* 4 inches; and using *b* as center with a radius of $1\frac{5}{8}$ inches draw the semicircle E–7–H. Draw lines from E to D and from C to H. D E 7 H C 12 D will be the plan of the bottom of the bath. In this case we assume that the flare between the top and bottom of the narrow end of the bath should be equal; therefore using *a* as center and with a radius equal to $1\frac{5}{8}$ inches draw the semicircle A W B. At the upper end of the bath the flare will be unequal; therefore from *b* measure a distance on line W 8 of 1 inch and obtain *c*, which use as center, and with a radius equal to 2 inches describe the arc F 8 G. Draw lines from F to A and from B to G; and A F 8 G B W A will be the plan of the top of the bath. Now project the side elevation from the plan as shown by the dotted lines, making the slant height from I to R $2\frac{1}{8}$ inches and from J to K $3\frac{1}{8}$ inches; draw a line

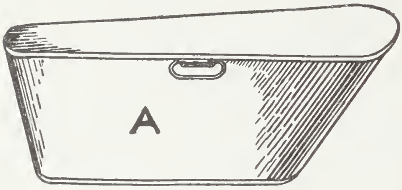


Fig. 26.

from K to R, and J K R I will be the side elevation of the bath tub. In constructing the bath in practice, seams are located at H G, F E,

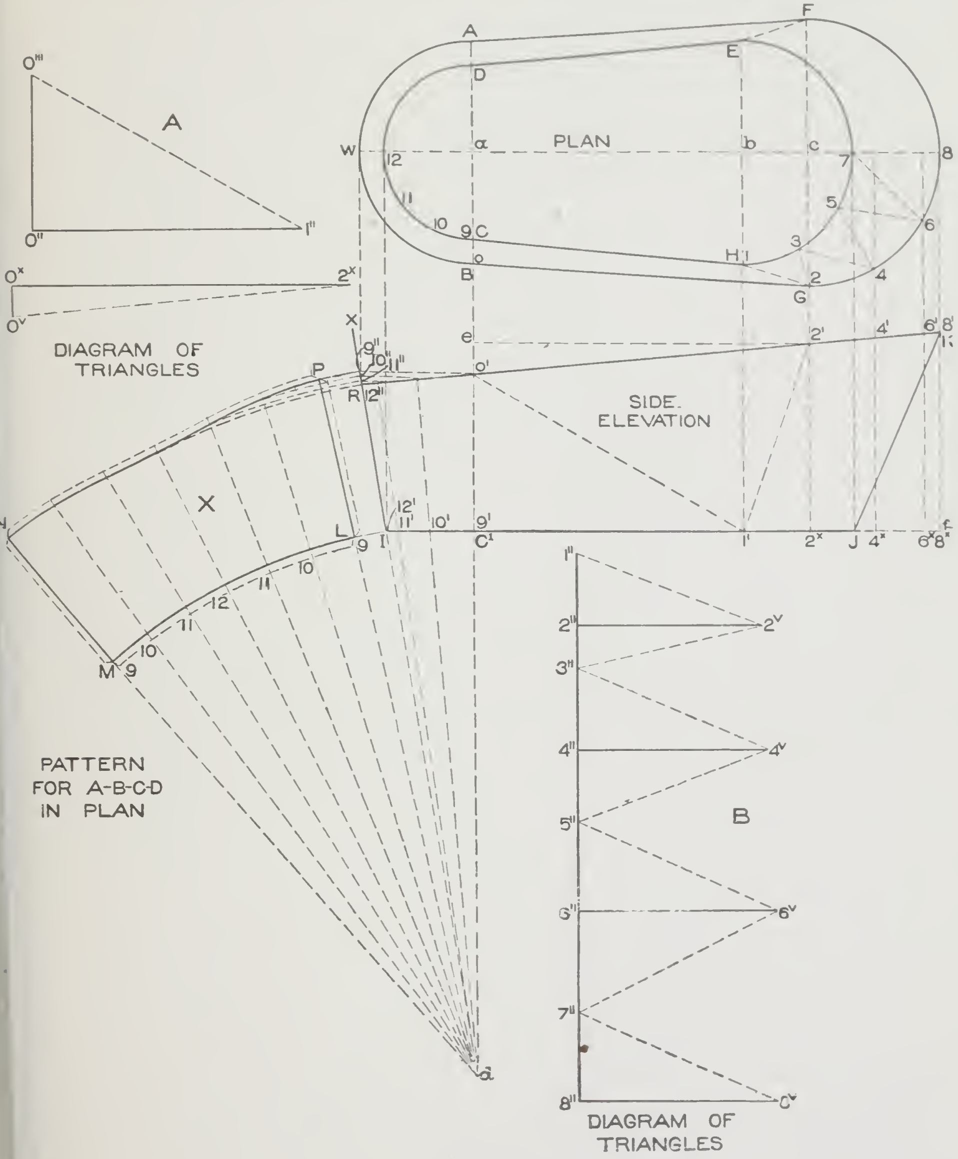


Fig. 27.

A D, and C B in plan, thus making the tub in four pieces

The lower end of the bath will be developed by the cone method as in the last two problems. From the center *a* drop a line indefinitely as shown. Extend the side R I of the side elevation until it meets the center line *a d* at *d*. Now divide the quarter circle 12–9 in plan into equal spaces as shown by the small figures 9, 10, 11, and 12, from which drop vertical lines (not shown) intersecting the bottom of the bath tub in elevation from 9′ to 12′. Then through these points from *d* draw lines intersecting the top line of the bath R K as shown, from which draw horizontal lines intersecting the side I–R extended as I X at points 9″ to 12″. Then using *d* as center and *d* I as radius, describe the arc I M, upon which place the stretchout of D 12 C in plan, as shown by similarly numbered points on L M. Through these points from *d* draw radial lines, which intersect by arcs drawn from similarly numbered intersections on I R extended, using *d* as center. Trace a line as shown, and L M N P will be the pattern for the lower end of the tub A B C D in plan. Laps should be allowed for wiring and seaming.

As the patterns for the upper end and sides will be developed by triangulation, diagrams of triangles must first be obtained for which proceed as follows: Divide both of the quarter circles H 7 and G 8 in plan into the same number of spaces as shown respectively from 1 to 7 and from 2 to 8. Connect these numbers by dotted lines as shown from 1 to 2, 2 to 3, 3 to 4, etc. From the various points 2, 4, 6, and 8 representing the top of the bath, drop lines meeting the base line J *f* in elevation at 2^x, 4^x, 6^x, and 8^x, and cutting the top line of the bath at 2′, 4′, 6′, and 8′. Then will the dotted lines in plan represent the bases of the triangles, which will be constructed, whose altitudes are equal to the various heights in elevation. Take the various distances 1 to 2, 2 to 3, 3 to 4, 4 to 5, etc., in plan up to 8, and place them on the vertical line 1″–8″ in (B) as shown from 1″ to 2″, 2″ to 3″, 3″ to 4″, 4″ to 5″, etc., up to 8″. For example, to obtain the true length of the line 6–7 in plan, remembering that the points having even numbers represent the top line of the bath and those having uneven numbers the base line, draw at right angles to 1″–8″ in (B), from 6″, a line equal in height to 6^x–6′ in elevation, and draw a line from 6^v to 7″ in (B), which is the length desired. For the true length of 6–5 in plan it is necessary only to take this distance place it from 6″ to 5″ in (B) and draw a line from 6^v to 5″. In this way each altitude answers for two triangles. In plan draw a line from 1 to 0. Then will two more triangles be necessary, one on the line 1–0, and the other on B G or 0–2. From 2′ in elevation draw a horizontal line, as 2′ *e*, intersecting the vertical line dropped from 0 at *e*. Now take the distances 0 1 and 0 2, and place them in (A) as shown by the horizontal lines 0″–1″ and 0^x–2^x respectively. At right angles to both lines at either end draw the vertical lines 0″–0‴ and 0^x–0^v equal in height respectively to C^1 0′ and *e* 0′ in elevation. Draw in (A) lines from 2^x to 0^v and from 1″ to 0‴, which are the desired lengths. Before proceeding with the pattern, a true section must be obtained on 2′–8′ in side elevation. Take the various distances 2′ to 8′ and place them on the line 2′–8′ in Fig. 28. At right angles to 2′–8′ and through the small figures draw lines as shown. Now measuring in each and every instance from the center line in plan in Fig. 27, take the various distances to points 2, 4, and 6 and place them on similarly numbered lines in Fig. 28, measuring in each case on either side of the line 2′–8′, thus obtaining the intersections 2–4–6. A line traced through these points will be the true section on 2′–8′ in elevation in Fig. 27.

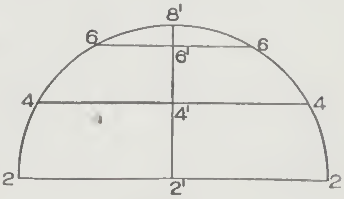


Fig. 28.

For the pattern for the upper end of the tub proceed as follows: Take the distance of 7″–8^v in (B) and place it on the vertical line 7–8 in Fig. 29. Then using 8 as center and with a radius equal to 8′–6 in Fig. 28, describe the arc 6 in Fig. 29, which intersect by an arc struck from 7 as center and with 7″–6^v in (B) in Fig. 27 as radius. Then using 7–5 in plan as radius, and 7 in Fig. 29 as center, describe the arc 5, which intersect by an arc struck from 6 as center and with 6^v–5″ in (B) in Fig. 27 as radius. Proceed in this manner, using alternately as radii first the divisions in Fig. 28, then the length of the slant lines in (B) in Fig. 27, the divisions on 7 H in plan, then again the slant lines in B, until the line 1–2 in Fig. 29 is obtained. Trace a line through points thus obtained, as shown by 2–8–7–1. Trace this opposite the line 8–7, as shown by 2′ 1′. Then will 2–8–2′–1′–7–1 be the desired pattern, to which laps must be allowed.

For the pattern for the side of the bath draw any line 9–1 in Fig. 30 equal to 9–1 in plan in Fig. 27. Now with a radius equal to 9–P in the pattern X and with 9 in Fig. 30 as a center, describe the arc 0, which intersect by an arc struck from 1 as center and with 1″–0‴ in (A) in Fig. 27 as radius. Now taking a radius equal to 0^{v}–2^{x} in (A) with 0 in Fig. 30 as center, describe the arc 2, which intersect by an arc struck from 1 as center, and with 1–2 in Fig. 29 as radius. Draw lines from corner to corner in Fig. 30, which gives the desired pattern, to which laps are added for seaming and wiring.

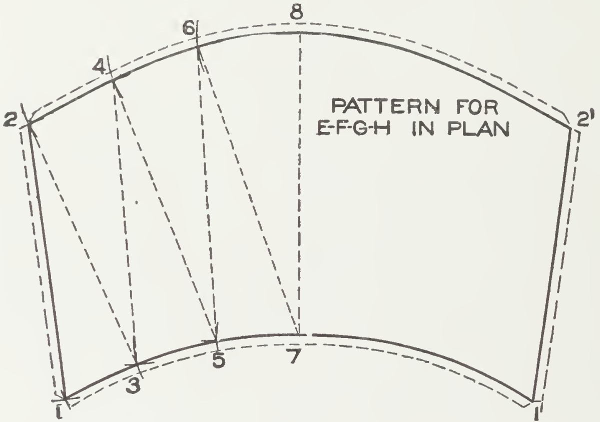


Fig. 29.

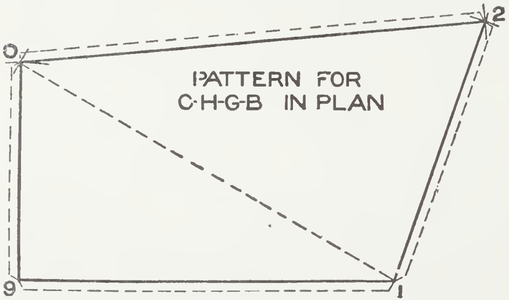


Fig. 30.

In Fig. 31 is shown a perspective view of a funnel strainer pail. These pails are usually made from IX bright tin, and the same principles as are used in the development of the pattern are applicable to similar forms, such as buckets, coal hods, chutes, etc. This problem presents an interesting study in triangulation, the principles of which have been explained in previous problems. First draw the center line C I in Fig. 32, at right angles to which draw H E and H F each equal to 1¼ inches. Make the vertical height H C 3¼ inches and C D 2 inches. Now make the vertical heights measuring from C G, to A, and to B respectively 1¼ inches, and 1½ inches. Make the horizontal distance from C to G 2¾ inches, the diameter from G to A 1⅜ inches, and from A to B ¾-inch, and draw a line from B to C. Connect points by lines; then will A B C D E F G be the side elevation of the pail. In its proper position below F E, with J as center, draw the plan K L M N. Also in its proper position draw the section on A G as O P R S. Now draw the rear elevation making G^1 U and G^1 V each equal to H E, and 1″ T and 1″–1′ each equal to C D. Project a line from B in side, intersecting the center line in rear at 4′. Then through the three points 1′ 4′ T draw the curve at pleasure, which in this case is struck from the center *a*. W Y X Z represents the opening on G A in side obtained as shown by the dotted lines but having no bearing on the patterns. Pails of this kind are usually made from two pieces, with seams at the sides, as in Fig. 31. The pattern then for the back shown by C D E H in side elevation in Fig. 32 will be obtained by the cone method, struck from the center I, the stretchout on E^1 E^2 in the pattern being obtained from the half plan. The pattern for C D E H is shown with lap and wire allowances by D^1 D^2 E^2 E^1 and needs no further explanation.

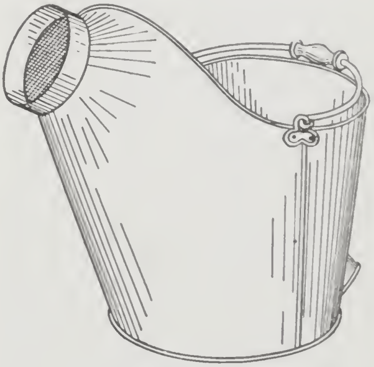


Fig. 31.

The front part of the pail shown by A B C H F G will be developed by triangulation, but before this can be done a true section must be obtained on B C, and a set of sections developed as follows: Divide one-half of 1′ 4′ T in rear elevation into equal parts as shown from 1′ to 4′, from which draw horizontal lines intersecting the line B C as shown. From these intersections lines are drawn at right angles to B C equal in length to similarly numbered lines in rear as 3′–3″, 2′–2″, and 1′–1″. Trace a line as shown, so that C 1‴ 2‴ 3‴ 4‴ will be the true half section on B C. To avoid a confusion of lines take a tracing of A B C H F G and place it as shown by similar letters in Fig. 33. Now take tracings of the half sections in Fig. 32, as H E D C, C 1‴ B, P O S, and the quarter plan N J M, and place them in Fig. 33 on similar lines on which they represent sections as shown respectively by H 9′ 8′ C, C 8 B, A 3 G, and F 9 H. Divide the half section

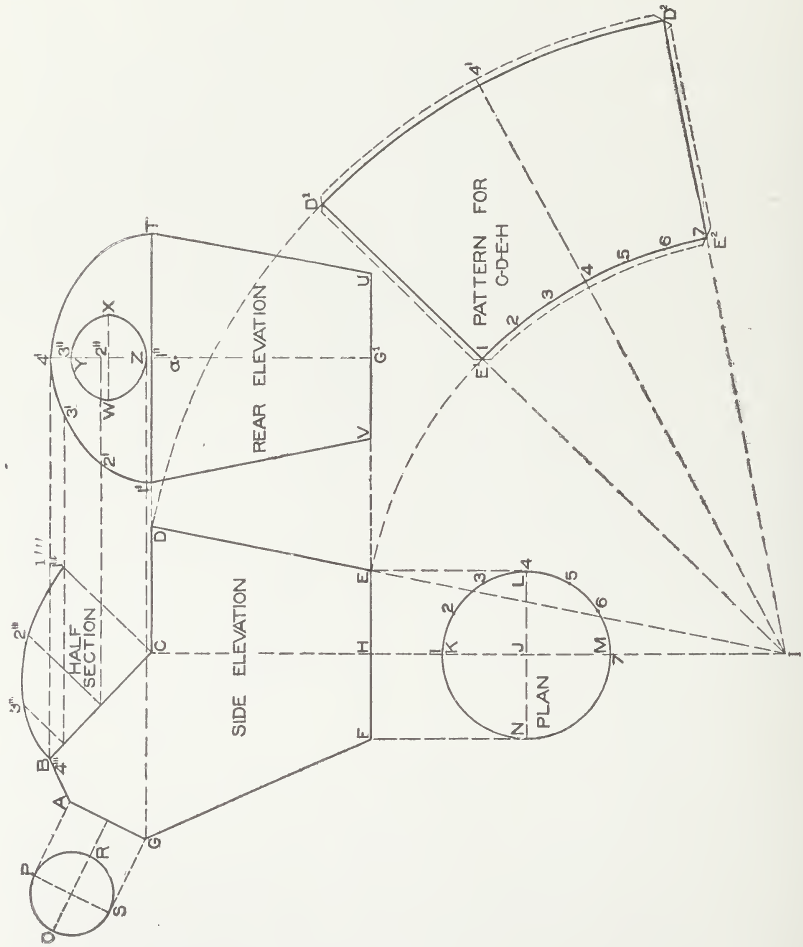


Fig. 32.

A 3 G into 6 equal parts as shown by the small figures 1 to 5. As this half section is divided into 6 parts, then must each of the sections B 8 C and F 9 H be divided into 3 parts as shown respectively from 6 to 8 and 9 to 11. As C 8′ and H 9′ are equal respectively to C 8 and H 9 they are numbered the same as shown.

Now at right angles to G A, B C, C H, and H F, and from the various intersections contained in the sections G 3 A, B 8 C, C 8′ 9′ H, and H 9 F, draw lines intersecting the base lines of the sections G A, B C, C H, and H F at points shown from 1′ to 11′. Now draw dotted lines from B to 5′ to 6′ to 4′ to 7′ to E to C, and then from H to E to 10′ to 2′, etc until all the points are connected as shown. These dotted lines represent the bases of the sections whose altitudes are equal to similar numbers in the various sections.

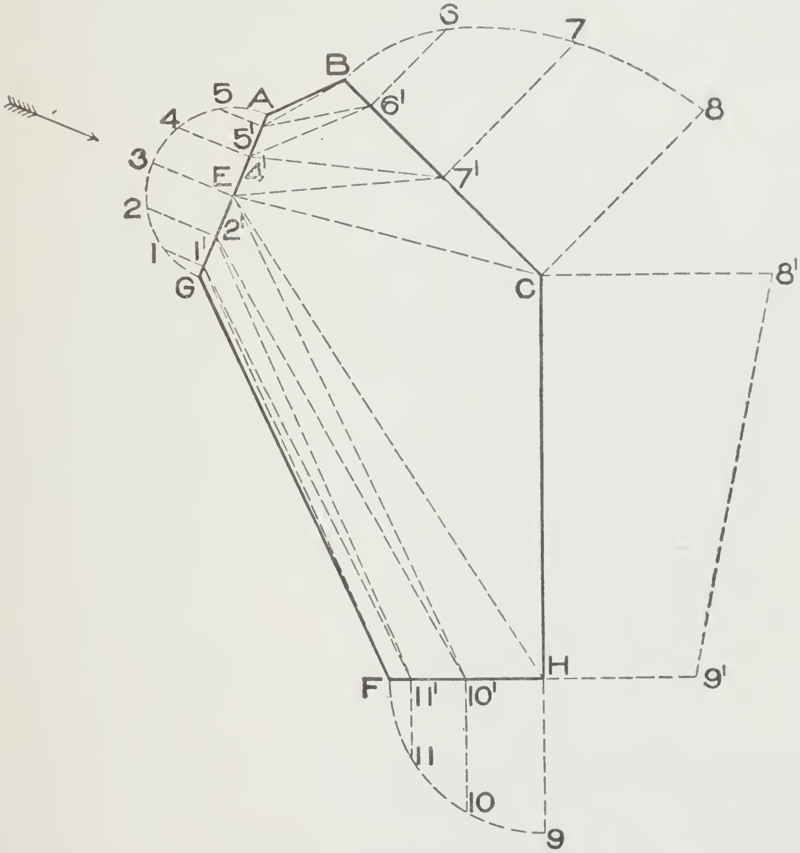


Fig. 33.

In order that the student may thoroughly understand this method of triangulation as well as similar methods that will follow in other problems, the model in Fig. 34 has been prepared, which shows a perspective of Fig. 33 with the sections bent up in their proper positions. This view is taken on the arrow line in Fig. 33, the letters and figures in both views being similar. For the true sections on the dotted lines in C E A B in Fig. 33, take the lengths of the dotted lines C E, E 7′, 7′ 4′, etc., and place them on the horizontal line in Fig. 35 as shown by similar letters and figures. From these small figures, at right angles to the horizontal line, erect the vertical heights C 8, E 3, 7′ 7, etc., equal to similar vertical heights in the sections in Fig. 33. Connect these points in Fig. 35 by dotted lines as shown, which are the desired true distances.

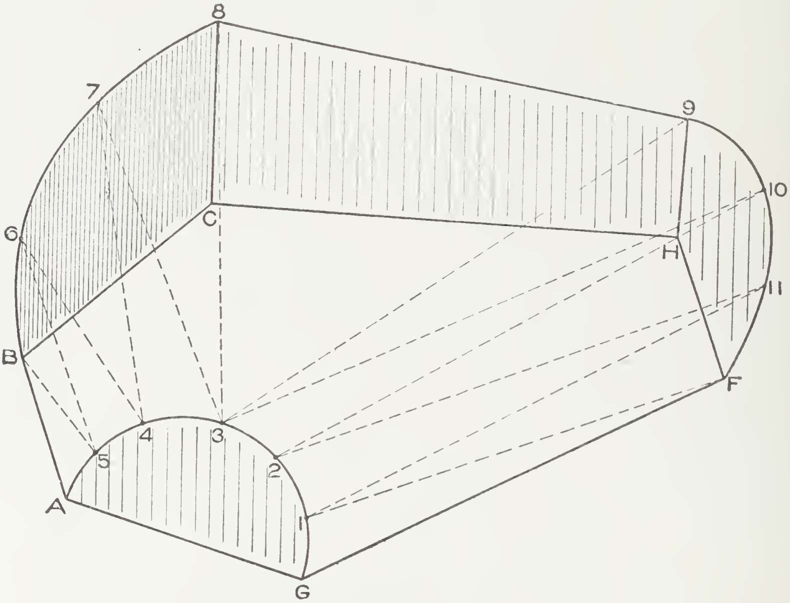


Fig. 34.

In Fig. 36 are shown the true sections on dotted lines in G E H F in Fig. 33, which are obtained in precisely the same manner, the only difference being that one section is placed inside of another in Fig. 36. For the pattern proceed as is shown in Fig. 37. Draw any vertical line as G F equal to G F in Fig. 33. With radius equal to G 1 and with G in Fig. 37 as center describe the arc 1, which intersect by an arc struck from F as center and with a radius equal to F 1 in Fig. 36. Now with F 11 in Fig. 33 as radius and F in Fig. 37 as center, describe the arc 11, which is intersected by an arc struck from 1 as center and with 1–11 in Fig 36 as radius. Proceed in this manner until the line 3–9 in Fig. 37 has been obtained. Then using 8′–9′ in Fig. 33 as radius and 9 in Fig. 37 as center, describe the arc 8, which is intersected by an arc struck from 3 as center and with 3–8 in Fig. 35 as radius. Now use alternately as radii, first the divisions in B 8 C in Fig. 33, then the length of the slant lines in Fig. 35, the divisions in E 3 A in Fig. 33, and again the distances in Fig. 35, until the line B A in Fig. 37 has been obtained, which is obtained from B A in Fig. 33. Trace a line through points thus obtained in Fig. 37 as shown by A B 8 9 F G A. Trace this half pattern opposite the line G F. Then will B A G A¹ B¹ 8¹ 9¹ F 9 8 be the pattern for the front half of the pail. If for any reason the pattern is desired in one piece, then trace one-half of D¹ D² E² E¹ in Fig. 32 on either side of the pattern in Fig. 37 as shown by the dotted lines 8′ D¹ E¹ 9¹ and 9 E D 8. Allow edges for wiring and seaming.

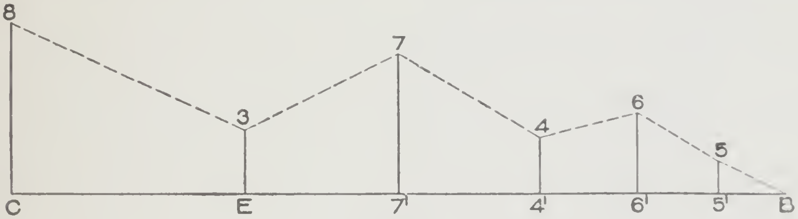


Fig. 35.

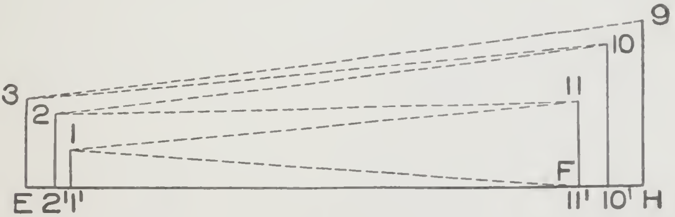


Fig. 36

Fig 38 shows the method for obtaining the pattern for an Emerson ventilator shown in Fig. 39.

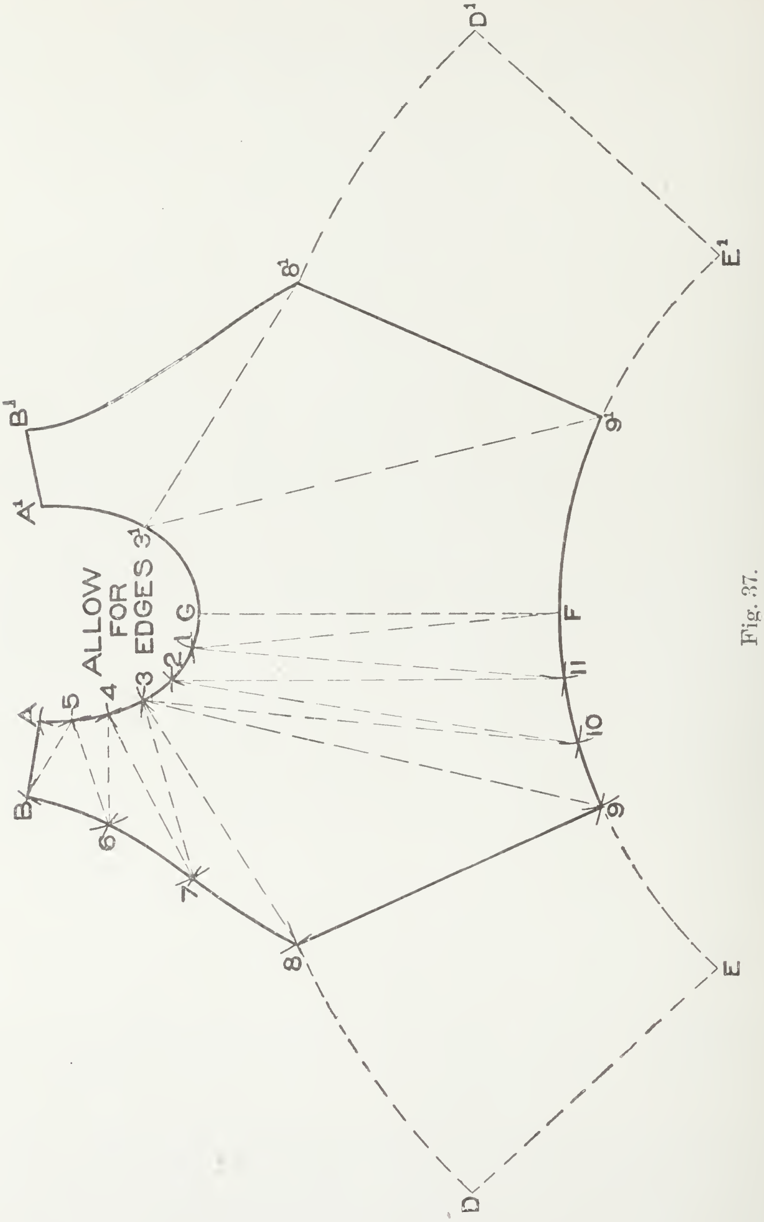


Fig. 37.

While the regular Emerson ventilator has a flat disc for a hood it is improved by placing a cone and deflector on the top as shown. To make the patterns, proceed as shown in Fig. 38. First draw the center line *a b*, on either side of which lay off

1½ inches, making the pipe A, 3 inches in diameter. The rule usually employed is to make the diameter of the lower flare and upper hood twice the diameter of the pipe. Therefore make the diameter of *s d* 6 inches. From *s* and *d*, draw a line at an angle of 45° to intersect the line of the pipe at *t* and *i*; this completes B. Measure 2 inches above the line *t i* and make *u m* the same diameter as *s d*. Draw the bevel of the deflector so that the apex will be ½ inch above the line *t i* and make the apex of the hood the same distance above *u m* as the lower apex is below it. Then draw lines as shown which complete C and D. Now with *c* as a center and radii equal to *c e* and *c d* draw the quarter circles *e f* and *d h* respectively, which represent the one-quarter pattern for the horizontal ring closing the bottom of the lower flare. For the pattern for the hood, use *l* as a center and *l m* as a radius. Now draw the arc *m m'*. Take the stretchout

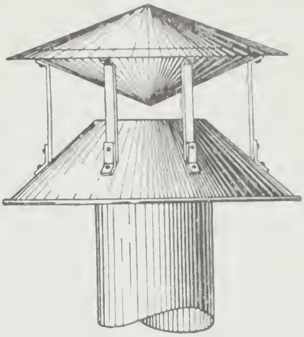


Fig. 39.

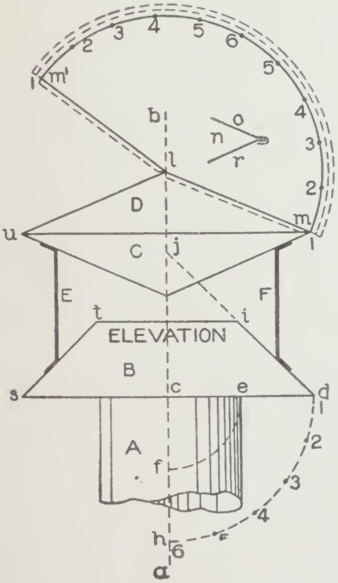


Fig. 38.

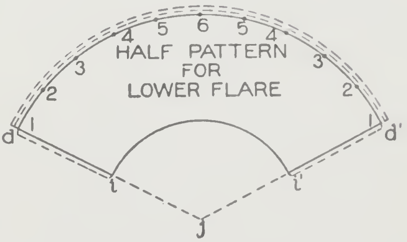


Fig. 40.

of the quarter circle 1 to 6 on $d h$, and place twice this amount on $m m'$ as shown from 1–6–1. Draw a line from 1 to l. Then m' 6 $m l$, will be the half pattern for the hood. As the deflector has the same bevel as the hood, the hood pattern will also answer for the deflector.

When seaming the hood and deflector together as shown at n, the hood o is double-seamed to the deflector at r, which allows the water to pass over; for this reason allow a double edge on the pattern for the hood as shown, while on the deflector but a single edge is required. Edges should also be allowed on $e d h f$.

For the pattern for the lower flare, extend the line $d i$ until it intersects the center line at j. Then with radii equal to $j i$ and $j d$ and with j in Fig. 40 as center describe the arcs $i i'$ and $d d'$. On one side as d draw a line to j. Then set off on the arc $d d'$ twice the number of spaces contained in $d h$ in Fig. 38 as shown in Fig. 40. Draw a line from d' to i and allow edges for seaming. Then $d d' i' i$ will be the half pattern for the lower flare.

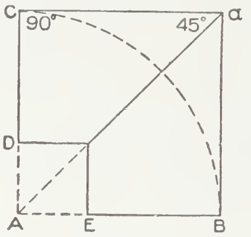


Fig. 41.

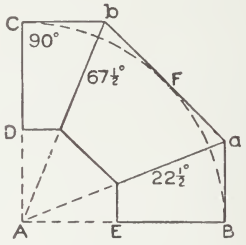


Fig. 42.

The braces or supports E and F, Fig. 38, are usually made of galvanized band iron bolted or riveted to hood and pipe. The hood D must be water tight or the water will leak into the deflector, from which it will drip from the apex inside the building.

Elbows. There is no other article in the sheet-metal worker's line, of which there are more made in practice than elbows. On this account rules will be given for constructing the rise of the miter line in elbows of any size or diameter, also for elbows whose sections are either oval, square or round, including tapering elbows Before taking up the method of obtaining the patterns, the rule will be given for obtaining the rise of the miter line for any size or number of pieces. No matter how many pieces an elbow has, they join together and form an angle of 90°. Thus when we speak of a two-pieced, three-pieced, four, five or six-pieced elbow, we understand that the right-angled elbow is made up of that number of pieces. Thus in Fig. 41 is shown a two-pieced elbow placed in the quadrant C B, which equals 90° and makes C A B a right angle. From A draw the miter line A *a* at an angle of 45° to the base line A B. Then parallel to A B and A C and tangent to the quadrant at C and B draw lines to intersect the miter line, as shown. Knowing the diameter of the pipe as C D or E B draw lines parallel to the arms of the pipe, as shown. Then C B E D will be a two-pieced elbow, whose miter line is an angle of 45°.

In a similar manner draw the quadrant B C, Fig. 42, in which it is desired to draw a three-pieced elbow. Now follow this simple rule, which is applicable for any number of pieces: Let the top piece of the elbow represent 1, also the lower piece 1, and for every piece between the top and bottom add 2. Thus in a three-pieced elbow:

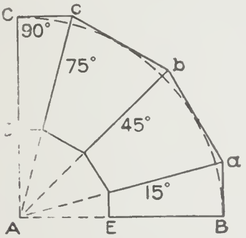


Fig. 43.

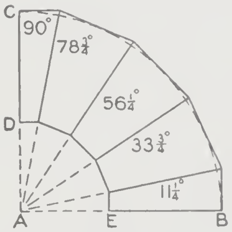


Fig. 44.

Top piece equals	1
Bottom piece equals	1
One piece between	2
Total equals	4

Now divide the quadrant of 90° by 4 which leaves 22½°. As one piece equals 22½°, draw the lower miter line A *a* at that angle to the base line A B. Then as the middle piece represents two by the above rule and equals 45°, add 45 to 22½ and draw the second miter line A *b*, at an angle of 67½° to the base line A B. Now tangent to the quadrant at C and B draw the vertical and horizontal lines shown, until they intersect the miter lines, from which intersections draw the middle line, which will be tangent to the quadrant at F. C D and B E show the diameters of the pipe, which are drawn parallel to the lines of the elbow shown.

Fig. 43 shows a four-pieced elbow, to which the same rule is applied. Thus the top and bottom piece equals 2 and the two middle pieces equal 4; total 6. Now divide the quadrant of 90° by 6. $\frac{90}{6} = 15$. Then the first miter line A *a* will equal 15°, the second A *b* 45°, the third A *c* 75°, and the vertical line A C 90°.

The last example is shown in Fig. 44, which shows a five-pieced elbow, in which the top and bottom pieces equal 2, the 3 middle pieces 6; total 8. Divide 90 by 8. $\frac{90}{8} = 11\frac{1}{4}$. Then the first miter line will equal $11\frac{1}{4}$°, the second $33\frac{3}{4}$°, the third $56\frac{1}{4}$°, and the fourth $78\frac{3}{4}$°. By using this method an elbow having any number of pieces may be laid out. When drawing these miter lines it is well to use the protractor shown in Fig. 45, which illustrates how to lay out a three-pieced elbow. From the center point A of the protractor draw lines through $22\frac{1}{2}$°, and $67\frac{1}{2}$°. Now set off A *a*, and the diameter of the pipe *a b*. Draw vertical lines from *a* and *b* to the miter line at *c* and *d*. Lay off similar distances from A to *a'* to *b'* and draw horizontal lines intersecting the $67\frac{1}{2}$° miter line at *c'* and *d'*. Then draw the lines *d d'* and *c c'* to complete the elbow. In practice, however, it is not necessary to draw out the entire view of the elbow; all that is required is the first miter line, as will be explained in the following problems.

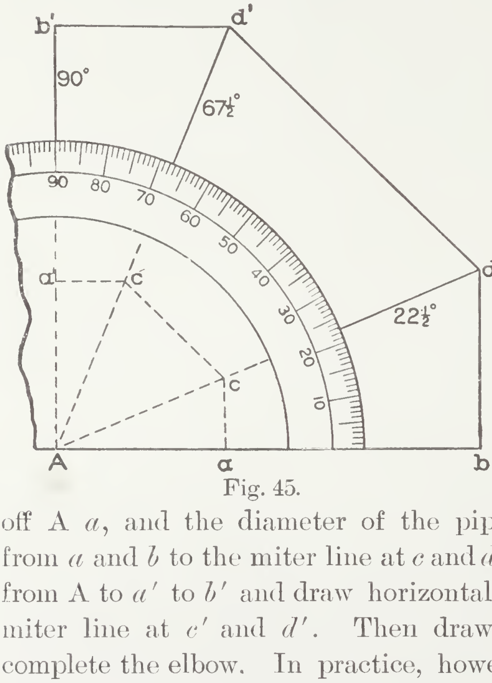


Fig. 45.

EXERCISES FOR PRACTICE.

1. Make the diameter of the pipe 1¾ inches and the distances from A to E 1½ inches in Figs. 41 to 44 inclusive.

To obtain the pattern for any elbow, using but the first miter line, proceed as follows: In Fig. 46 let A and B represent respectively a two- and three-pieced elbow for which patterns are desired. First draw a section of the elbow as shown at A in Fig. 47 which is a circle 3 inches in diameter; divide the lower half into equal spaces and number the points of division 1 to 7. Now follow the rule previously given: The top and bottom piece equals 2; then

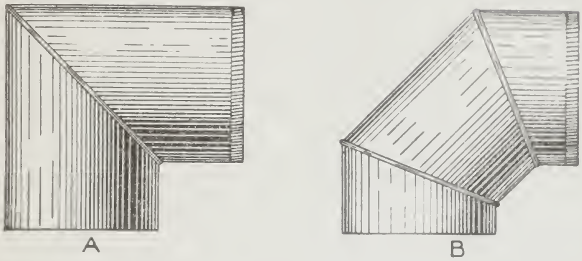


Fig. 46.

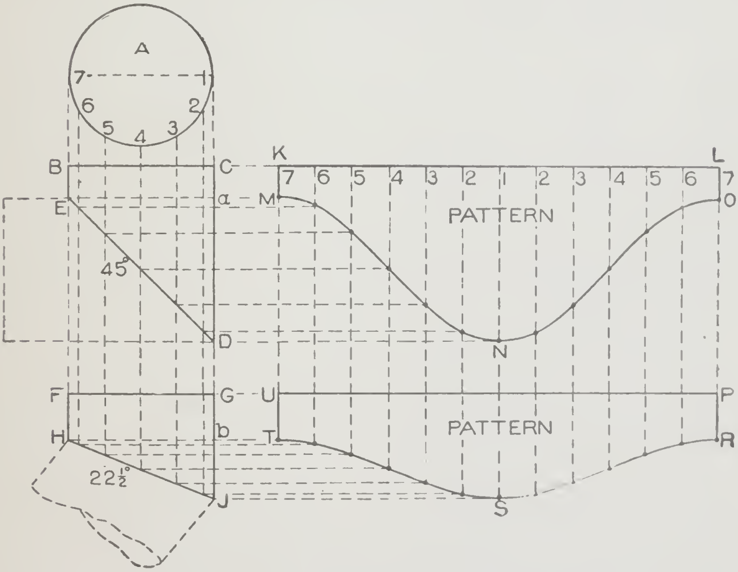


Fig. 47.

for a two-pieced elbow divide 90 by 2. In its proper position below the section A draw B C D E making E D 45°. From the various points of intersection in A drop vertical lines intersecting E D as shown. In line with B C draw K L upon which place twice the number of spaces contained in the section A as shown by similar figures on K L; from these points drop perpendiculars to intersect

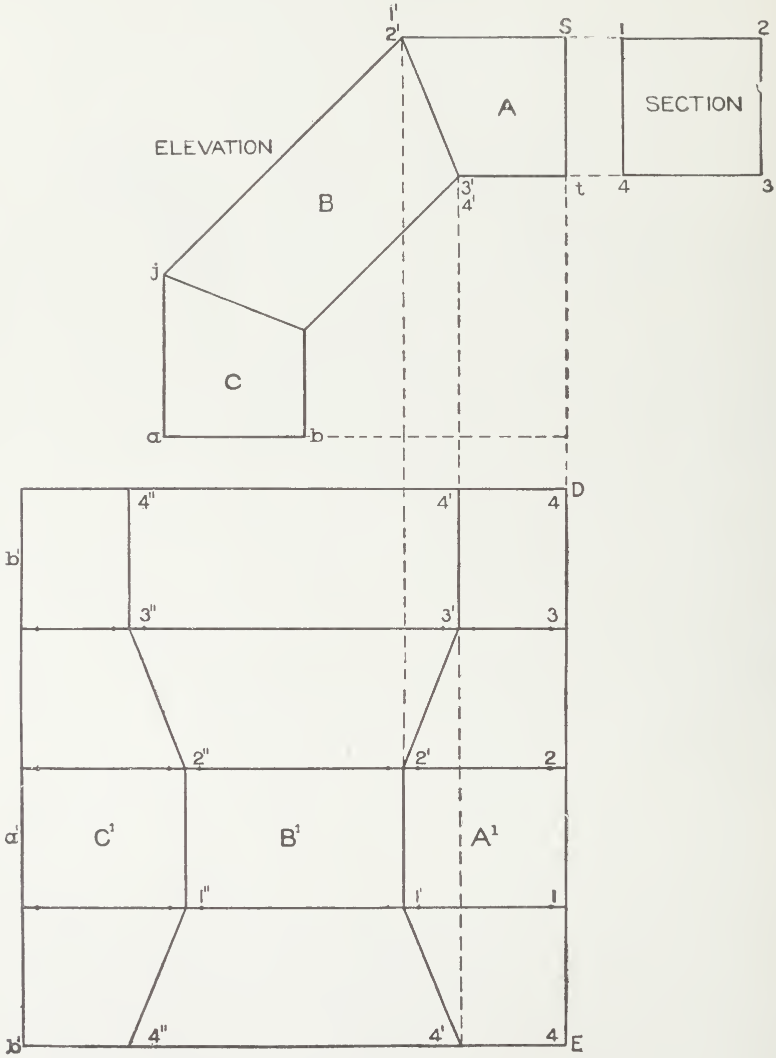


Fig. 48.

with lines drawn from similar intersections on E D, parallel to K L. Trace a line through points shown; then K L O N M will be the pattern. To this laps must be allowed for seaming.

Now to obtain the pattern for a three-pieced elbow, follow the rule. Top and bottom pieces equal 2, one middle piece equals 2; total 4. $\frac{90}{4} = 22\frac{1}{2}$. Therefore in line with the section A below the two-pieced elbow draw F G J H, making H J at an angle of $22\frac{1}{2}°$ to the line H b. Proceed as above using the same stretchout lines; then U P R S T will be the desired pattern. It should be understood that when the protractor is used for obtaining the angle as shown in Fig. 45, the heights a c and b d measured from the horizontal line form the basis for obtaining the heights of the middle pieces, inasmuch as they represent one-half the distance; for that reason the middle pieces count 2 when using the rule. Therefore, the distances F H and G J (Fig. 47), represent one-half of the center piece and U T S R P one-half the pattern for the center piece of a three-pieced elbow.

Fig. 48 shows how the patterns are laid into one another, to prevent waste of metal when cutting. In this example we have a three-pieced elbow whose section is 2 × 2 inches. It is to be laid out in a quadrant whose radius is 5 inches. Use the same principles for square section as for round; number the corners of the section 1 to 4. In line with S t draw D E upon which place the stretchout of the square section as shown by similar numbers on D E; from which draw horizontal lines which intersect lines drawn parallel to D E from the intersections $1'$ $2'$ and $3'$ $4'$ in A in elevation, thus obtaining similar points in the pattern. Then A^1 will be the pattern for A in elevation. For the pattern f r B simply take the distance from $2'$ to j and place it on the line 4 $4'$ extended in the pattern on either side as shown by $4'$ $4''$ on both sides. Now reverse the cut $4'$ $2'$ $4'$ and obtain $4''$ $2''$ $4''$. By measurement it will be found that $4'$ $4''$ is twice the length of $2'$ 2 as explained in connection with Figs. 45 and 47. Make the distance from $1''$ to a' the same as j to a in C and draw the vertical line b' b' intersecting the lines 4 $4''$ extended on both sides. Then A^1, B^1. and C^1 will be the patterns in one piece minus the edges for seaming which must be allowed between these cuts; this would of course make the lengths b′ 4″, 4″ 4′ and 4′ 4 as much longer as the laps would necessitate.

This method of cutting elbows in one piece, from one square is applicable to either round, oval or square sections.

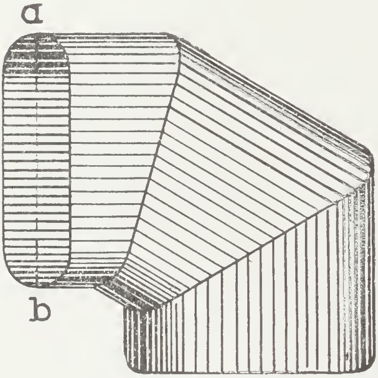


Fig. 49.

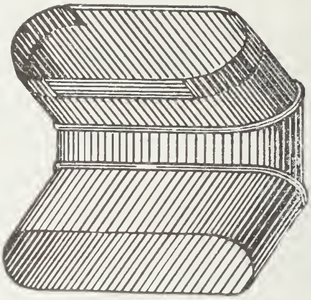


Fig. 50.

In Figs. 49 and 50 are shown three-pieced elbows such as are used in furnace-pipe work and are usually made from bright tin. Note the difference in the position of the sections of the two elbows. In Fig. 49 *a b* is in a vertical position, while in Fig. 50 it is in a horizontal position. In obtaining the patterns the same rule is employed as in previous problems, care being taken when developing the patterns for Fig. 49 that the section be placed as in Fig. 51 at A; and when developing the patterns for Fig. 50, that the section be placed as shown at A in Fig. 52.

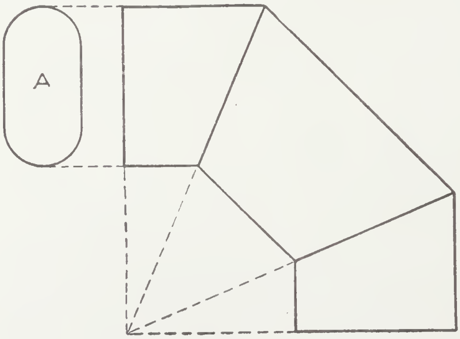


Fig. 51.

Fig. 53 shows a tapering two-pieced elbow, round in section. The method here shown is short and while not strictly accurate, gives good results. It has been shown in previous problems on Intersections and Developments that an oblique section through the opposite sides of a cone is a true ellipse. Bearing this in mind it is evident that if the frustum of the cone H I O N, Fig. 54, were a solid and cut obliquely by the plane J K and the several parts placed side by side, both would present true ellipses of exactly the same size, and if the two parts were placed together again turning the upper piece half-way around as shown by J W M K, the edges of the two pieces from J to K would exactly coincide. Taking advantage of this fact, it is necessary only to ascertain the angle of the line J K, to produce the required angle, between the two pieces of the elbow, both of which have an equal flare. The angle of the miter line, or the line which cuts the cone in two parts, must be found accurately so that when joined together an elbow will be formed having the desired angle on the line of its axis.

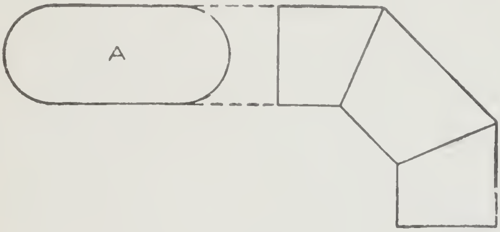


Fig. 52.

Therefore draw any vertical line as A B. With C as a center describe the plan of the desired diameter as shown by E D F B. At right angles to A B draw the bottom line of the elbow H I equal to E F, or in this case, 3 inches. Measuring from the line H I on the line A B the height of the frustum is 5 inches. Through X′ draw the upper diameter O N, $1\frac{1}{2}$ inches. Extend the contour lines of the frustum until they intersect the center line at L. Divide the half plan E D F into a number of equal parts as shown; from these points erect lines intersecting the base line H I from which draw lines to the apex L. As the elbow is to be in two pieces, and the axis at right angles, draw the angle T R S,

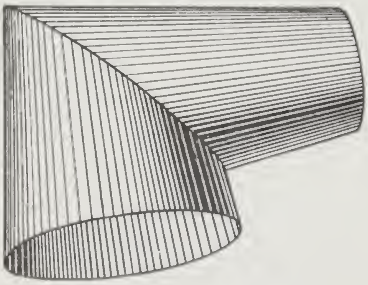


Fig. 53.

bisect it at U and draw the line R V. No matter what the angle of the elbow, use this method. Now establish the point J at some convenient point on the cone, and from J, parallel to R V, draw the miter line J K intersecting the radial lines drawn through the cone; from these points and at right angles to the center line A B draw lines intersecting the side of the cone J H from 1 to 7. If it is

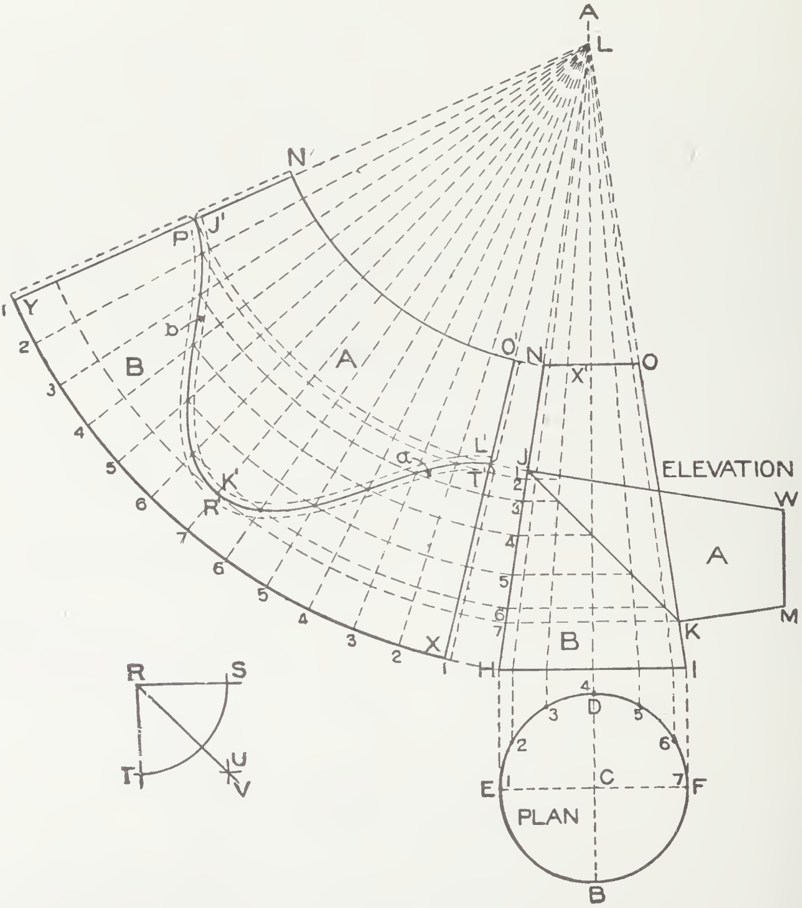


Fig. 54.

desired to know how the side of the tapering elbow would look, take a tracing of N O K J, reverse it and place it as shown by J W M K.

For the pattern proceed as follows: With L as a center and L H as a radius describe the arc 1 1. Starting from 1 set off on this arc twice the stretchout of 1 4 7 in plan, as shown by similar figures on 1 1, from which draw radial lines to the apex L. Again using L as center with radii equal to L N, L 1, L 2 to L 7, draw arcs as shown intersecting radial lines having similar numbers. Through these intersections draw the line J′ L′. Then O′ N′ J′ K′ L′ or A will be the pattern for the upper arm (A) in elevation, and P′ R′ T′ X Y or B the pattern for the lower arm (B) in elevation.

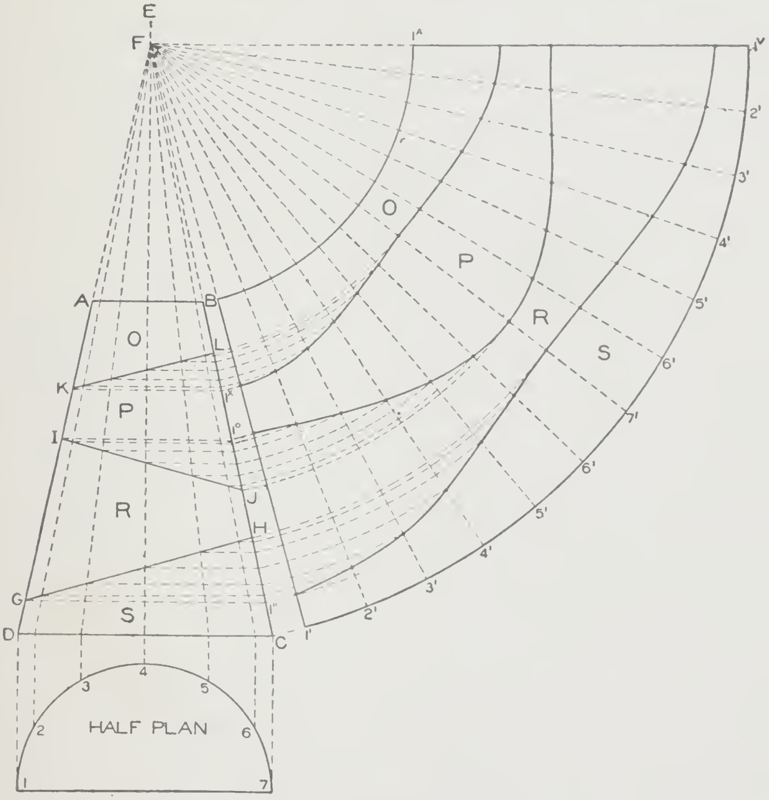


Fig. 55.

The pattern should be developed full size in practice and then pricked from the paper on to the sheet metal, drawing the two patterns as far apart as to admit allowing an edge to A at *a*; also an edge at *b* to B for seaming.

When a pattern is to contain more than two pieces the method of constructing the miter lines in the elevation of the cone is slightly different as shown in Fig. 55. Assume the bottom to be 3 inches in diameter and the top $1\frac{1}{4}$ inches. Let the vertical height be 4 inches. In this problem, as in the preceding, the various pieces necessary to form the elbow are cut from one cone whose dimensions must be determined from the dimensions of the required elbow. The first step is to determine the miter lines, which can be done the same as if regular pieced elbows were being developed. As the elbow is to consist of four pieces in 90°, follow the rule given in connection with elbow drafting. The top and bottom piece equal 2; the two middle pieces equal 4; total 6. $\frac{90}{6} = 15$. Lay off A B C D according to the dimensions given, and draw the half plan below D C; divide it into equal parts as shown. From the points of division erect perpendiculars intersecting D C, from which draw lines meeting the center line E 4 at F.

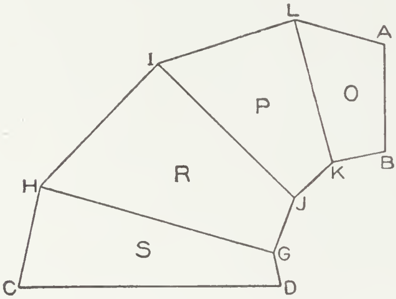


Fig. 56.

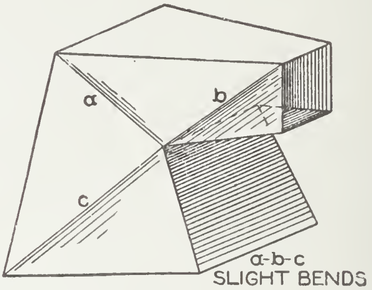


Fig. 57.

We assume that the amount of rise and projection of the elbow are not specified, excepting that the lines of axis will be at right angles. Knowing the angle of the miter line, it becomes a matter of judgment upon the part of the pattern draftsman, what length shall be given to each of the pieces composing the elbow. Therefore establish the points G, I and K, making D G, G I, I K and K A, $\frac{1}{2}$, $1\frac{7}{8}$, $\frac{3}{4}$ and 1 inch respectively. From G, I and K draw the horizontal lines G $1''$, I 1° and K 1^x. To each of these lines draw the lines G H, I J and K L respectively at an angle of 15° intersecting the radial lines in the cone as shown. From these intersections draw horizontal lines cutting the side of the cone. Then using F as a center, obtain the various patterns O, P, R and S in the manner already explained.

In Fig. 56 is shown a side view of the elbow, resulting from preceding operations; while it can be drawn from dimensions obtained in Fig. 55, it would be impossible to draw it without first having these dimensions.

In Fig. 57 is shown a perspective view of a tapering square elbow of square section in two pieces. This elbow may have any given taper. This problem will be developed by triangulation and parallel lines; it is an interesting study in projections as well as in developments. First draw the elevation of the elbow in Fig. 58 making 1–6 equal to $3\frac{1}{2}$ inches, the vertical height 1–2, $4\frac{1}{2}$ inches, and 6–5, $2\frac{1}{2}$ inches; the projection between 1 and 2 should be $\frac{5}{8}$ inch and between 5 and 6, $\frac{3}{8}$ inch. Make the horizontal distance from 5 to 4, 2 inches, and the rise at 4 from the horizontal line $\frac{1}{4}$ inch, and the vertical distance from 4 to 3, $1\frac{1}{4}$ inches. Then draw a line from 3 to 2 to complete the elevation.

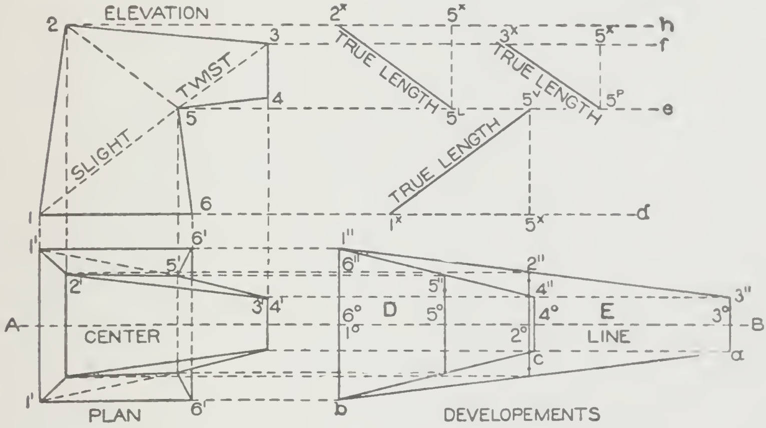


Fig. 58.

In its proper position below the line 1–6, draw the plan on that line, as shown by 1′ 1′ 6′ 6′. Through this line draw the center line A B. As the elbow should have a true taper from 1 to 3 and from 4 to 6, we may develop the patterns for the top and bottom pieces first and then from these construct the plan. Therefore, take the distances from 1 to 2 to 3 and from 4 to 5 to 6 in elevation and place them on the line A B in plan as shown respectively from 1° to 2° to 3° and from 4° to 5° to 6°; through these points draw vertical lines as shown. While the full developments

E and D are shown we shall deal with but one-half in the explanation which follows. As the elbow is to have the same taper on either side, take the half distance of the bottom of the elbow 1–6 and place it as shown from 1°–6° to 1″–6″, and the half width of the top of the elbow 3–4 and place it as shown from 3° to 3″ and 4° to 4″. Then draw lines from 3″ to 1″ intersecting the bend 2° at 2″, and a line from 4″ to 6″ intersecting the bend 5° at 5″. Trace these points on the opposite side of the line A B. Then 1″ 3″ *a b* will be the pattern for the top of the elbow and 6″ 4″ *c b* the pattern for the bottom. From these various points of intersection draw horizontal lines to the plan, and intersect them by lines drawn from similarly numbered points in the elevation at right angles to A B in plan. Draw lines through the points thus obtained in plan as shown by 1′, 2′, 3′, 4′, 5′ and 6′ which will represent the half plan view. For the completed plan, trace these lines opposite the line A B as shown. It will be noticed that the line 3–4 in elevation is perpendicular as shown by 3′ 4′ in plan while the points 2′ and 5′ project from it, showing that the piece 2–3–4–5 in elevation must be slightly twisted along the line 5–3 when forming the elbow. Similarly slight bends will be required along the lines 1–5 and 5–2.

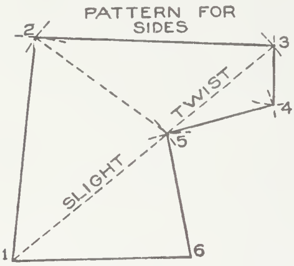


Fig. 59.

It will now be necessary to obtain the true lengths or a diagram of triangles on the lines 1–5, 5–2 and 5–3. Connect similar numbers in plan as shown from 1′ to 5′, 5′ to 2′ and 5′ to 3′, the last two lines being already shown. From similar points in elevation draw horizontal lines as shown by 2–*h*, 3–*f*, 5–*e* and 6–*d*. Take the distances from 1′ to 5′, 5′ to 2′ and 5′ to 3′ in plan and place them on one of the lines having a similar number in elevation, as shown respectively by 1^{x} 5^{x}, 5^{x} 2^{x} and 5^{x} 3^{x}. From the points marked 5^{x} draw vertical lines intersecting the horizontal line drawn from 5 at 5^{v}, 5^{L} and 5^{P} respectively. Now draw the true lengths 1^{x} 5^{v}, 2^{x} 5^{L}, and 3^{x} 5^{P}. For the pattern draw any line as 1–6 in Fig. 59 equal to 1–6 in Fig. 58. Now with 6″ 5″ in D as a radius and 6 in Fig. 59 as a center, describe the arc 5 which is intersected by an arc struck from 1 as a center and the true length

1^x 5^v in Fig. 58 as radius. Then using the true length 5^L 2^x as radius and 5 in Fig. 59 as center, describe the arc 2, which is intersected by an arc struck from 1 as center and $1''$ $2''$ in E in Fig. 58 as radius. Using the true length 5^p 3^x as radius and 5 in Fig. 59 as center, describe the arc 3, and intersect it by an arc struck from 2 as center and $2''$ $3''$ in E in Fig. 58 as a radius. Now with $5''$ $4''$ in D as a radius and 5 in Fig. 59 as a center, describe the arc 4, and intersect it by an arc struck from 3 as center and 3–4 in the elevation in Fig. 58 as a radius. Draw lines from point to point in Fig. 59 to complete the pattern. Laps should be allowed on all patterns, for seaming. Slight bends will take place as shown on the pattern, also as is shown by *a b* and *c* in Fig. 57. If the joint is to be on the line 2–5 in elevation in Fig. 58, the necessary pieces can be joined together.

In Fig. 60 is shown a perspective view of a five-piece tapering elbow, having a round base and an elliptical top. This form is generally known as a *ship ventilator*. The principles shown in this problem are applicable to any form or shape no matter what the respective profiles may be at the base or top. The first step is to draw a correct side view of the elbow as shown in Fig. 61. The outline A B C D E F can be drawn at pleasure, but for practice, dimensions are given. First draw the vertical line A F equal to $4\frac{1}{2}$ inches. On the same line extend measure down $1\frac{1}{4}$ inches to *f* and draw the horizontal line H B. From *f* set off a distance of $1\frac{1}{4}$ inches at G, and using G as a center and G F as a radius describe the arc F E intersecting H B at E, from which draw the vertical line E D equal to 1 inch. Draw D C equal to $1\frac{3}{4}$ inches, then draw C B. From B lay off $5\frac{3}{4}$ inches, and using this point (H) as a center and H B as a radius describe the arc B A. The portion shown B E D C is a straight piece of pipe whose section is shown by I J K L. Now divide the two arcs B A and E F into the same number of parts that the elbow is to have pieces (in this case four) and draw the lines of joint or miter lines as shown by U V, etc.

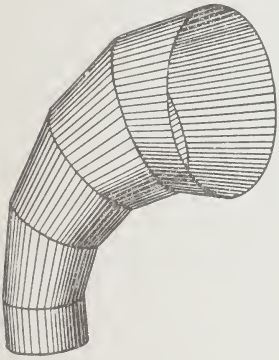


Fig. 60.

Bisect each one of the joint lines and obtain the points *a b c d* and *e*. Then A B C D E F will be the side view.

The patterns will be developed by triangulation, but before this can be done, true sections must be obtained on all of the lines in side elevation. The true sections on the lines B E and C D are shown by I J K L. The length of the sections are shown by the joint lines, but the width must be obtained from a front outline of the elbow, which is constructed as follows: In its proper relation to the side elevation, draw the center line M R upon which draw

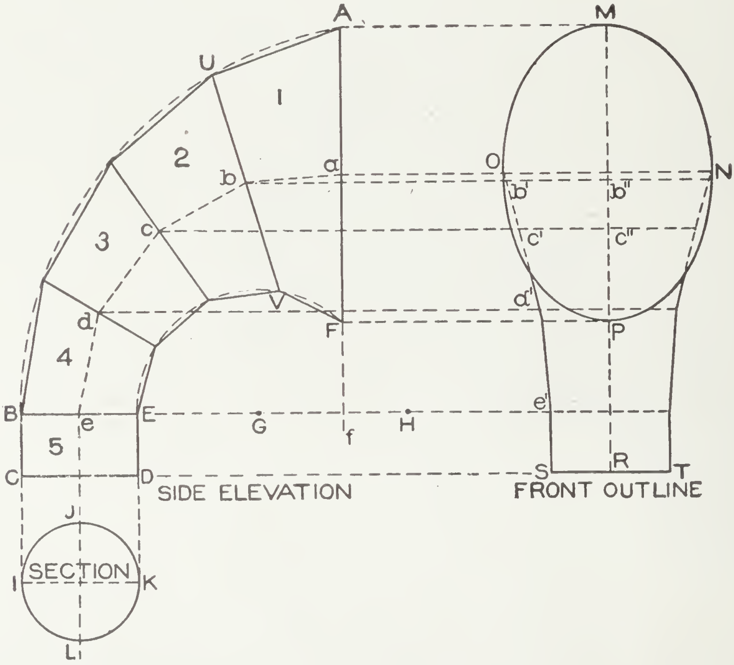


Fig. 61.

the ellipse M N O P (by methods already given in Mechanical Drawing) which represents the section on A F in side. Take half the diameter I K in section and place it on either side of the center line M R as R T or R S. Then draw the outline O S and T N in a convenient location. While this line is drawn at will, it should be understood that when once drawn, it becomes a fixed line. Now from the various intersections *a b c d* and *e* in the side elevation, draw lines through and intersecting the front outline as shown on one side by O, *b′*, *c′*, *d′* and *e′*. Then these distances will represent the widths of the sections shown by similar letters in side. For example, the method will be shown for obtaining the true section on U V, and the pattern for piece 1 in side elevation. To avoid a confusion of lines take a tracing of A F V U and place it as shown by 1, 13, 12, O in Fig. 62. On 1–13 place the half profile M N P of Fig. 61. Bisect O–12 in Fig. 62 and obtain the point 6; at a right angle to O–12 from 6 draw the line 6 6′ equal to *b′ b″* in front outline in Fig. 61. Then through the three points O, 6′ and 12 in Fig. 62, draw the semi-ellipse, which will represent the half section on U V. The other sections on the joint lines in side elevation are obtained in the same manner.

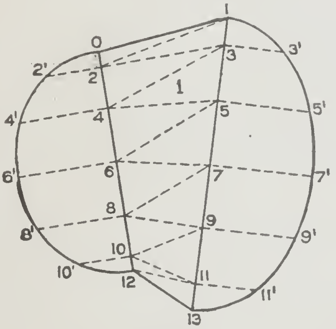


Fig. 62.

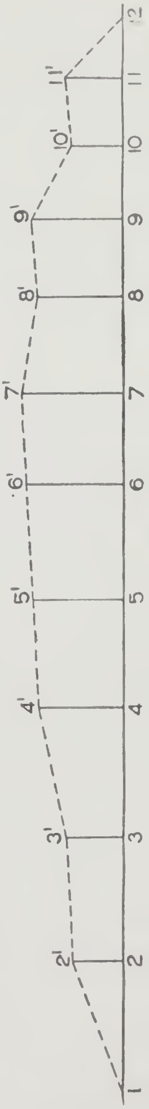


Fig. 63.

If the sections were required for piece 2 in side it would be necessary to use only O 6′ 12 in Fig. 62 and place it on U V in Fig. 61, and on a perpendicular line erected from *c*, place the width *c′ c″* shown in front and through the three points obtained again draw the semi-elliptical profile or section. Now divide the two half sections (Fig. 62) into equal parts as shown by the small figures, from which at right angles to 1–13 and O–12 draw lines intersecting these base lines from 1–13. Connect opposite points as 1 to 2 to 3 to 4 to 5, etc., to 12. Then these lines will represent the bases of sections whose altitudes are equal to the heights in the half section. For these heights proceed as follows:

Take the various lengths from 1 to 2, 2 to 3, 3 to 4, 4 to 5, etc., to 11 to 12 and place them on the horizontal line in Fig. 63 as shown by similar figures; from these points erect vertical lines equal in height to similar figures, in the half section in Fig. 62 as shown by similar figures in Fig. 63. For example: Take the distance from 7 to 8 in Fig. 62 and place it as shown from 7 to 8 in Fig. 63 and erect vertical lines 7–7′, and 8–8′ equal to 7–7′ and 8–8′ in Fig. 62. Draw a line from 7′ to 8′ in Fig. 63 which is the true length on 7–8 in Fig. 62. For the pattern take the distance of 1–O and place it as shown by 1–O in Fig. 64. Now using O as a center and O 2′ in Fig. 62 as a radius, describe the arc 2 in Fig. 64 and intersect it by an arc struck from 1 as a center with 1–2′ in Fig. 63 as a radius. Now with 1–3′ in Fig. 62 as a radius and 1 in Fig. 64 as a center, describe the arc 3, and intersect it by an arc struck from 2 as center and 2′–3′ in Fig. 63 as a radius. Proceed thus, using alternately as radii, first the divisions in O–6′–12 in Fig. 62, then the proper line in Fig. 63, the divisions in 1–7′–13 in Fig. 62 and again the proper line in Fig. 63, until the line 12–13 in Fig. 64 is obtained, which equals 12–13 in Fig. 62. In this manner all of the sections are obtained, to which laps must be allowed for wiring and seaming.

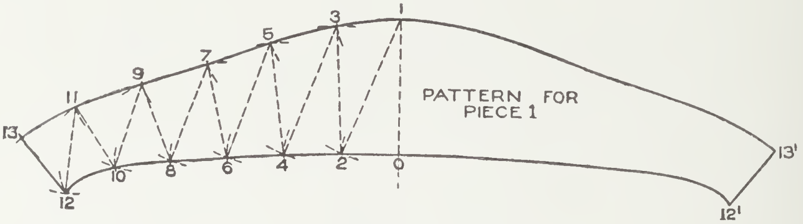


Fig. 64.

TABLES.

The following tables will be found convenient for the Sheet-Metal Worker:

TABLES	PAGE.
Weight of Cast Iron, Wrought Iron, Copper, Lead, Brass and Zinc	62
Sheet Copper	63
Sheet Zinc	64
Standard Gauge for Sheet Iron and Steel	65
Weights of Flat Rolled Iron	66-71
Square and Round Iron Bars	72-73
Angles and Tees	74

WEIGHT OF A SQUARE FOOT OF CAST AND WROUGHT IRON, COPPER, LEAD, BRASS AND ZINC.

FROM $\frac{1}{16}$ INCH TO ONE INCH IN THICKNESS.

THICKNESS.	CAST IRON.	WROUGHT IRON.	COPPER.	LEAD.	BRASS.	ZINC.
Inch.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
1-16	2.346	2.517	2.89	3.691	2.675	2.34
1-8	4.693	5.035	5.781	7.382	5.35	4.68
3-16	7.039	7.552	8.672	11.074	8.025	7.02
1-4	9.386	10.07	11.562	14.765	10.7	9.36
5-16	11.733	12.588	14.453	18.456	13.375	11.7
3-8	14.079	15.106	17.344	22.148	16.05	14.04
7-16	16.426	17.623	20.234	25.839	18.725	16.34
1-2	18.773	20.141	23.125	29.53	21.4	18.72
9-16	21.119	22.659	26.016	33.222	24.075	
5-8	23.466	25.176	28.906	36.923	26.75	
11-16	25.812	27.694	31.797	40.604	29.425	
3-4	28.159	30.211	34.688	44.296	32.1	
13-16	30.505	32.729	37.578	47.987		
7-8	32.852	35.247	40.469	51.678		
15-16	35.199	37.764	43.359	55.37		
1	37.545	40.282	46.25	59.061		

NOTE.—The wrought iron and the copper weights are those of hard-rolled plates.

SHEET COPPER.

Official table adopted by the Association of Copper Manufacturers of the United States. Rolled copper has specific gravity of 8.93. One cubic foot weighs 558.125 pounds. One square foot, one inch thick, weighs 46.51 pounds.

Stubs' Gauge, (nearest number).	Thickness in decimal parts of an inch.	Ounces per sq. ft.	Sheets 14"x48" Weight in pounds.	Sheets 24"x 48" Weight in pounds.	Sheets 30"x 60" Weight in pounds.	Sheets 36"x 72" Weight in pounds.	Sheets 48"x 72" Weight in pounds.
35	.00537	4	1.16	2	3.12	4.50	6
33	.00806	6	1.75	3	4.68	6.75	9
31	.0107	8	2.33	4	6.25	9	12
29	.0134	10	2.91	5	7.81	11.25	15
27	.0161	12	3.50	6	9.37	13.50	18
26	.0188	14	4.08	7	10.93	15.75	21
24	.0215	16	4.66	8	12.50	18	24
23	.0242	18	5.25	9	14.06	20.25	27
22	.0269	20	5.83	10	15.62	22.50	30
21	.0322	24	7	12	18.75	27	36
19	.0430	32	9.33	16	25	36	48
18	.0538	40	11.66	20	31.25	45	60
16	.0645	48	14	24	37.50	54	72
15	.0754	56	16.33	28	43.75	63	84
14	.0860	64	18.66	32	50	72	96
13	.095	70	....	35	55	79	105
12	.109	81	....	40½	63	91	122
11	.120	89	....	44½	70	100	134
10	.134	100	....	50	78	112	150
9	.148	110	....	55	86	124	165
8	.165	123	....	61	96	138	184
7	.180	134	....	67	105	151	201
6	.203	151	....	75½	118	170	227
5	.220	164	....	82	128	184	246
4	.238	177	....	88½	138	199	266
3	.259	193	....	96	151	217	289
2	.284	211	....	105½	165	238	317
1	.300	223	....	111½	174	251	335
0	.340	253	....	126½	198	285	380

SHEET ZINC.

Numbers		4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
Weight per sq. foot		.30	.37	.45	.52	.60	.67	.75	.90	1.05	1.20	1.35	1.50	1.68	1.87	2.06	2.25	2.62	3.00	3.37
Approximate thickness in inches		.008	.010	.012	.014	.016	.018	.020	.024	.028	.032	.036	.040	.045	.050	.055	.060	.070	.080	.090
Size of sheet.	Sq. ft. per sheet.	APPROXIMATE WEIGHT PER SHEET.																		
24 x 84 in.	14.	4.2	5.2	6.3	7.3	8.4	9.4	10.5	12.6	14.7	16.8	18.9	21.	23.5	26.2	28.9	31.5	36.7	42.	47.2
26 x 84	15.2	4.6	5.6	6.9	7.9	9.1	10.2	11.4	13.7	16.	18.3	20.5	22.8	25.6	28.4	31.3	34.2	39.9	45.6	51.2
28 x 84	16.3	4.9	6.	7.4	8.5	9.8	10.9	12.2	14.7	17.1	19.6	22.	24.5	27.4	30.5	33.6	36.7	42.7	48.9	54.9
30 x 84	17.5	5.3	6.5	7.9	9.1	10.5	11.8	13.2	15.8	18.4	21.	23.6	26.2	29.4	32.8	36.1	39.4	45.8	52.5	59.
32 x 84	18.7	5.6	6.9	8.4	9.7	11.2	12.6	14.1	16.9	19.7	22.5	25.3	32.8	31.4	35.	38.5	42.	49.	56.1	63.
34 x 84	19.9	6.0	7.4	9.	10.4	12.	13.4	15.	18.	20.9	23.9	26.9	29.9	33.4	37.2	41.	44.8	52.2	59.7	67.
36 x 84	21.	6.3	7.8	9.5	10.9	12.6	14.1	15.8	18.9	22.	25.2	28.4	31.5	35.3	39.3	43.3	47.2	55.	63.	70.8
36 x 96	24.	7.2	8.9	10.8	12.5	14.4	16.1	18.	21.6	25.2	28.8	32.4	36.	40.3	44.9	49.5	54.	62.8	72.	80.9
36 x 108	27.	8.1	10.	12.2	14.1	16.2	18.1	20.3	24.3	28.4	32.4	36.5	40.5	45.4	50.5	55.6	60.7	70.7	81.	91.
40 x 84	23.4	7.	8.7	10.6	12.2	14.1	15.7	17.6	21.	24.6	28.1	31.6	35.1	39.3	43.8	48.2	52.6	61.3	70.2	78.8
40 x 96	26.8	8.	9.9	12.1	14.	16.1	18.	20.1	24.1	28.1	32.2	36.2	40.2	45.	50.1	55.2	60.3	70.2	80.4	90.3
44 x 84	25.7	7.7	9.5	11.6	13.4	15.4	17.2	19.3	23.1	27.	30.8	34.7	38.6	43.2	48.1	53.	57.8	67.4	77.1	86.6
46 x 90	28.7	8.6	10.6	12.9	14.9	17.2	19.2	21.5	25.8	30.1	34.4	38.7	43.	48.2	53.7	59.1	64.6	75.2	86.1	96.7
48 x 84	28.	8.4	10.4	12.6	14.6	16.8	18.8	21.	25.2	29.4	33.6	37.8	42.	47.	52.4	57.7	63.	73.4	84.	94.4
48 x 96	32.	9.6	11.9	14.4	16.7	19.2	21.5	24.	28.8	33.6	38.4	43.2	48.	53.8	59.9	65.9	72.	83.9	96.	107.8
50 x 108	37.5	11.3	13.9	16.9	19.5	22.5	25.1	28.2	33.8	39.3	45.	50.7	56.3	63.	70.1	77.3	84.4	98.3	112.5	126.4
52 x 84	30.4	9.1	11.3	13.7	15.8	18.3	20.4	22.8	27.4	31.9	36.5	41.	45.6	51.	56.9	62.6	68.4	79.6	91.2	102.5

Casks average about 600 pounds each. No. 4 to No. 17. Boxes average about 500 pounds. No. 18 and heavier.

UNITED STATES STANDARD GAUGE FOR SHEET AND PLATE IRON AND STEEL

COPY [Public—No. 137]

An act establishing a standard gauge for sheet and plate iron and steel.

Be it enacted by the Senate and House of Representatives of the United States of America in Congress assembled, That for the purpose of securing uniformity the following is established as the only standard gauge for sheet and plate iron and steel in the United States of America, namely:

Number of Gauge	THICKNESS		WEIGHT		Number of Gauge
	Approximate thickness in fractions of an inch	Approximate thickness in decimal parts of an inch	Weight per square foot in OUNCES avoirdupois	Weight per square foot in POUNDS avoirdupois	
0000000	1-2	.5	320	20.	0000000
000000	15-32	.46875	300	18.75	000000
00000	7-16	.4375	280	17.5	00000
0000	13-32	.40625	260	16.25	0000
000	3-8	.375	240	15.	000
00	11-32	.34375	220	13.75	00
0	5-16	.3125	200	12.5	0
1	9-32	.28125	180	11.25	1
2	17-64	.265625	170	10.625	2
3	1-4	.25	160	10.	3
4	15-64	.234375	150	9.375	4
5	7-32	.21875	140	8.75	5
6	13-64	.203125	130	8.125	6
7	3-16	.1875	120	7.5	7
8	11-64	.171875	110	6.875	8
9	5-32	.15625	100	6.25	9
10	9-64	.140625	90	5.625	10
11	1-8	.125	80	5.	11
12	7-64	.109375	70	4.375	12
13	3-32	.09375	60	3.75	13
14	5-64	.078125	50	3.125	14
15	9-128	.0703125	45	2.8125	15
16	1-16	.0625	40	2.5	16
17	9-160	.05625	36	2.25	17
18	1-20	.05	32	2.	18
19	7-160	.04375	28	1.75	19
20	3-80	.0375	24	1.5	20
21	11-320	.034375	22	1.375	21
22	1-32	.03125	20	1.25	22
23	9-320	.028125	18	1.125	23
24	1-40	.025	16	1.	24
25	7-320	.021875	14	.875	25
26	3-160	.01875	12	.75	26
27	11-640	.0171875	11	.6875	27
28	1-64	.015625	10	.625	28
29	9-640	.0140625	9	.5625	29
30	1-80	.0125	8	.5	30
31	7-640	.0109375	7	.4375	31
32	13-1280	.01015625	6½	.40625	32
33	3-320	009375	6	.375	33
34	11-1280	.00859375	5½	.34375	34
35	5-640	.0078125	5	.3125	35
36	9-1280	.00703125	4½	.28125	36
37	17-2560	.0066406	4¼	.265625	37
38	1-160	.00625	4	.25	38

And on and after July first, eighteen hundred and ninety-three, the same and no other shall be used in determining duties and taxes levied by the United States of America on sheet and plate iron and steel. But this act shall not be construed to increase duties upon any articles which may be imported.

SEC. 2. That the Secretary of the Treasury is authorized and required to prepare suitable standards in accordance herewith.

SEC. 3. That in the practical use and application of the standard gauge hereby established a variation of two and one-half per cent either way may be allowed.

Approved, March 3, 1893.

WEIGHTS OF FLAT ROLLED IRON PER LINEAR FOOT.

Iron weighing 480 pounds per cubic foot.

Thickness in Inches.	1″	1¼″	1½″	1¾″	2″	2¼″	2½″	2¾″	12″
1/16	.208	.260	.313	.365	.417	.469	.521	.573	2.50
1/8	.417	.521	.625	.729	.833	.938	1.04	1.15	5.00
3/16	.625	.781	.938	1.09	1.25	1.41	1.56	1.72	7.50
1/4	.833	1.04	1.25	1.46	1.67	1.88	2.08	2.29	10.00
5/16	1.04	1.30	1.56	1:82	2.08	2.34	2.60	2.86	12.50
3/8	1.25	1.56	1.88	2.19	2.50	2.81	3.13	3.44	15.00
7/16	1.46	1.82	2.19	2.55	2.92	3.28	3.65	4.01	17.50
1/2	1.67	2.08	2.50	2.92	3.33	3.75	4.17	4.58	20.00
9/16	1.88	2.34	2.81	3.28	3.75	4.22	4.69	5.16	22.50
5/8	2.08	2.60	3.13	3.65	4.17	4.69	5.21	5.73	25.00
11/16	2.29	2.86	3.44	4.01	4.58	5.16	5.73	6.30	27.50
3/4	2.50	3.13	3.75	4.38	5.00	5.63	6.25	6.88	30.00
13/16	2.71	3.39	4.06	4.74	5.42	6.09	6.77	7.45	32.50
7/8	2.92	3.65	4.38	5.10	5.83	6.56	7.29	8.02	35.00
15/16	3.13	3.91	4.69	5.47	6.25	7.03	7.81	8.59	37.50
1	3.33	4.17	5.00	5.83	6.67	7.50	8.33	9.17	40.00
1 1/16	3.54	4.43	5.31	6.20	7.08	7.97	8.85	9.74	42.50
1 1/8	3.75	4.69	5.63	6.56	7.50	8.44	9.38	10.31	45.00
1 3/16	3.96	4.95	5.94	6.93	7.92	8.91	9.90	10.89	47.50
1 1/4	4.17	5.21	6.25	7.29	8.33	9.38	10.42	11.46	50.00
1 5/16	4.37	5.47	6.56	7.66	8.75	9.84	10.94	12.03	52.50
1 3/8	4.58	5.73	6.88	8.02	9.17	10.31	11.46	12.60	55.00
1 7/16	4.79	5.99	7.19	8.39	9.58	10.78	11.98	13.18	57.50
1 1/2	5.00	6.25	7.50	8.75	10.00	11.25	12.50	13.75	60.00
1 9/16	5.21	6.51	7.81	9.11	10.42	11.72	13.02	14.32	62.50
1 5/8	5.42	6.77	8.13	9.48	10.83	12.19	13.54	14.90	65.00
1 11/16	5.63	7.03	8.44	9.84	11.25	12.66	14.06	15.47	67.50
1 3/4	5.83	7.29	8.75	10.21	11.67	13.13	14.58	16.04	70.00
1 13/16	6.04	7.55	9.06	10.57	12.08	13.59	15.10	16.61	72.50
1 7/8	6.25	7.81	9.38	10.94	12.50	14.06	15.63	17.19	75.00
1 15/16	6.46	8.07	9.69	11.30	12.92	14.53	16.15	17.76	77.50
2	6.67	8.33	10.00	11.67	13.33	15.00	16.67	18.33	80.00

WEIGHTS OF FLAT ROLLED IRON PER LINEAR FOOT.

(Continued)

Thickness in Inches.	3"	3¼"	3½"	3¾"	4"	4¼"	4½"	4¾"	12"
$\frac{1}{16}$	.625	.677	.729	.781	.833	.885	.938	.990	2.50
$\frac{1}{8}$	1.25	1.35	1.46	1.56	1.67	1.77	1.88	1.98	5.00
$\frac{3}{16}$	1.88	2.03	2.19	2.34	2.50	2.66	2.81	2.97	7.50
$\frac{1}{4}$	2.50	2.71	2.92	3.13	3.33	3.54	3.75	3.96	10.00
$\frac{5}{16}$	3.13	3.39	3.65	3.91	4.17	4.43	4.69	4.95	12.50
$\frac{3}{8}$	3.75	4.06	4.38	4.69	5.00	5.31	5.63	5.94	15.00
$\frac{7}{16}$	4.38	4.74	5.10	5.47	5.83	6.20	6.56	6.93	17.50
$\frac{1}{2}$	5.00	5.42	5.83	6.25	6.67	7.08	7.50	7.92	20.00
$\frac{9}{16}$	5.63	6.09	6.56	7.03	7.50	7.97	8.44	8.91	22.50
$\frac{5}{8}$	6.25	6.77	7.29	7.81	8.33	8.85	9.38	9.90	25.00
$\frac{11}{16}$	6.88	7.45	8.02	8.59	9.17	9.74	10.31	10.89	27.50
$\frac{3}{4}$	7.50	8.13	8.75	9.38	10.00	10.63	11.25	11.88	30.00
$\frac{13}{16}$	8.13	8.80	9.48	10.16	10.83	11.51	12.19	12.86	32.50
$\frac{7}{8}$	8.75	9.48	10.21	10.94	11.67	12.40	13.13	13.85	35.00
$\frac{15}{16}$	9.38	10.16	10.94	11.72	12.50	13.28	14.06	14.84	37.50
1	10.00	10.83	11.67	12.50	13.33	14.17	15.00	15.83	40.00
$1\frac{1}{16}$	10.63	11.51	12.40	13.28	14.17	15.05	15.94	16.82	42.50
$1\frac{1}{8}$	11.25	12.19	13.13	14.06	15.00	15.94	16.88	17.81	45.00
$1\frac{3}{16}$	11.88	12.86	13.85	14.84	15.83	16.82	17.81	18.80	47.50
$1\frac{1}{4}$	12.50	13.54	14.58	15.63	16.67	17.71	18.75	19.79	50.00
$1\frac{5}{16}$	13.13	14.22	15.31	16.41	17.50	18.59	19.69	20.78	52.50
$1\frac{3}{8}$	13.75	14.90	16.04	17.19	18.33	19.48	20.63	21.77	55.00
$1\frac{7}{16}$	14.38	15.57	16.77	17.97	19.17	20.36	21.56	22.76	57.50
$1\frac{1}{2}$	15.00	16.25	17.50	18.75	20.00	21.25	22.50	23.75	60.00
$1\frac{9}{16}$	15.63	16.93	18.23	19.53	20.83	22.14	23.44	24.74	62.50
$1\frac{5}{8}$	16.25	17.60	18.96	20.31	21.67	23.02	24.38	25.73	65.00
$1\frac{11}{16}$	16.88	18.28	19.69	21.09	22.50	23.91	25.31	26.72	67.50
$1\frac{3}{4}$	17.50	18.96	20.42	21.88	23.33	24.79	26.25	27.71	70.00
$1\frac{13}{16}$	18.13	19.64	21.15	22.66	24.17	25.68	27.19	28.70	72.50
$1\frac{7}{8}$	18.75	20.31	21.88	23.44	25.00	26.56	28.13	29.69	75.00
$1\frac{15}{16}$	19.38	20.99	22.60	24.22	25.83	27.45	29.06	30.68	77.50
2	20.00	21.67	23.33	25.00	26.67	28.33	30.00	31.67	80.00

WEIGHTS OF FLAT ROLLED IRON PER LINEAR FOOT.

(Continued)

Thickness in Inches.	5″	5¼″	5½″	5¾″	6″	6¼″	6½″	6¾″	12″
$\frac{1}{16}$	1.04	1.09	1.15	1.20	1.25	1.30	1.35	1.41	2.50
$\frac{1}{8}$	2.08	2.19	2.29	2.40	2.50	2.60	2.71	2.81	5.00
$\frac{3}{16}$	3.13	3.28	3.44	3.59	3.75	3.91	4.06	4.22	7.50
$\frac{1}{4}$	4.17	4.38	4.58	4.79	5.00	5.21	5.42	5.63	10.00
$\frac{5}{16}$	5.21	5.47	5.73	5.99	6.25	6.51	6.77	7.03	12.50
$\frac{3}{8}$	6.25	6.56	6.88	7.19	7.50	7.81	8.13	8.44	15.00
$\frac{7}{16}$	7.29	7.66	8.02	8.39	8.75	9.11	9.48	9.84	17.50
$\frac{1}{2}$	8.33	8.75	9.17	9.58	10.00	10.42	10.83	11.25	20.00
$\frac{9}{16}$	9.38	9.84	10.31	10.78	11.25	11.72	12.19	12.66	22.50
$\frac{5}{8}$	10.42	10.94	11.46	11.98	12.50	13.02	13.54	14.06	25.00
$\frac{11}{16}$	11.46	12.03	12.60	13.18	13.75	14.32	14.90	15.47	27.50
$\frac{3}{4}$	12.50	13.13	13.75	14.38	15.00	15.63	16.25	16.88	30.00
$\frac{13}{16}$	13.54	14.22	14.90	15.57	16.25	16.93	17.60	18.28	32.50
$\frac{7}{8}$	14.58	15.31	16.04	16.77	17.50	18.23	18.96	19.69	35.00
$\frac{15}{16}$	15.63	16.41	17.19	17.97	18.75	19.53	20.31	21.09	37.50
1	16.67	17.50	18.33	19.17	20.00	20.83	21.67	22.50	40.00
$1\frac{1}{16}$	17.71	18.59	19.48	20.36	21.25	22.14	23.02	23.91	42.50
$1\frac{1}{8}$	18.75	19.69	20.63	21.56	22.50	23.44	24.38	25.31	45.00
$1\frac{3}{16}$	19.79	20.78	21.77	22.76	23.75	24.74	25.73	26.72	47.50
$1\frac{1}{4}$	20.83	21.88	22.92	23.96	25.00	26.04	27.08	28.13	50.00
$1\frac{5}{16}$	21.88	22.97	24.06	25.16	26.25	27.34	28.44	29.53	52.50
$1\frac{3}{8}$	22.92	24.06	25.21	26.35	27.50	28.65	29.79	30.94	55.00
$1\frac{7}{16}$	23.96	25.16	26.35	27.55	28.75	29.95	31.15	32.34	57.50
$1\frac{1}{2}$	25.00	26.25	27.50	28.75	30.00	31.25	32.50	33.75	60.00
$1\frac{9}{16}$	26.04	27.34	28.65	29.95	31.25	32.55	33.85	35.16	62.50
$1\frac{5}{8}$	27.08	28.44	29.79	31.15	32.50	33.85	35.21	36.56	65.00
$1\frac{11}{16}$	28.13	29.53	30.94	32.34	33.75	35.16	36.56	37.97	67.50
$1\frac{3}{4}$	29.17	30.63	32.08	33.54	35.00	36.46	37.92	39.38	70.00
$1\frac{13}{16}$	30.21	31.72	33.23	34.74	36.25	37.76	39.27	40.78	72.50
$1\frac{7}{8}$	31.25	32.81	34.38	35.94	37.50	39.06	40.63	42.19	75.00
$1\frac{15}{16}$	32.29	33.91	35.52	37.14	38.75	40.36	41.98	43.59	77.50
2	33.33	35.00	36.67	38.33	40.00	41.67	43.33	45.00	80.00

WEIGHTS OF FLAT ROLLED IRON PER LINEAR FOOT.

(Continued)

Thickness in Inches.	7″	7¼″	7½″	7¾″	8″	8¼″	8½″	8¾″	12″
$\frac{1}{16}$	1.46	1.51	1.56	1.61	1.67	1.72	1.77	1.82	2.50
$\frac{1}{8}$	2.92	3.02	3.13	3.23	3.33	3.44	3.54	3.65	5.00
$\frac{3}{16}$	4.38	4.53	4.69	4.84	5.00	5.16	5.31	5.47	7.50
$\frac{1}{4}$	5.83	6.04	6.25	6.46	6.67	6.88	7.08	7.29	10.00
$\frac{5}{16}$	7.29	7.55	7.81	8.07	8.33	8.59	8.85	9.11	12.50
$\frac{3}{8}$	8.75	9.06	9.38	9.69	10.00	10.31	10.63	10.94	15.00
$\frac{7}{16}$	10.21	10.57	10.94	11.30	11.67	12.03	12.40	12.76	17.50
$\frac{1}{2}$	11.67	12.08	12.50	12.92	13.33	13.75	14.17	14.58	20.00
$\frac{9}{16}$	13.13	13.59	14.06	14.53	15.00	15.47	15.94	16.41	22.50
$\frac{5}{8}$	14.58	15.10	15.63	16.15	16.67	17.19	17.71	18.23	25.00
$\frac{11}{16}$	16.04	16.61	17.19	17.76	18.33	18.91	19.48	20.05	27.50
$\frac{3}{4}$	17.50	18.13	18.75	19.38	20.00	20.63	21.25	21.88	30.00
$\frac{13}{16}$	18.96	19.64	20.31	20.99	21.67	22.34	23.02	23.70	32.50
$\frac{7}{8}$	20.42	21.15	21.88	22.60	23.33	24.06	24.79	25.52	35.00
$\frac{15}{16}$	21.88	22.66	23.44	24.22	25.00	25.78	26.56	27.34	37.50
1	23.33	24.17	25.00	25.83	26.67	27.50	28.33	29.17	40.00
$1\frac{1}{16}$	24.79	25.68	26.56	27.45	28.33	29.22	30.10	30.99	42.50
$1\frac{1}{8}$	26.25	27.19	28.13	29.06	30.00	30.94	31.88	32.81	45.00
$1\frac{3}{16}$	27.71	28.70	29.69	30.68	31.67	32.66	33.65	34.64	47.50
$1\frac{1}{4}$	29.17	30.21	31.25	32.29	33.33	34.38	35.42	36.46	50.00
$1\frac{5}{16}$	30.62	31.72	32.81	33.91	35.00	36.09	37.19	38.28	52.50
$1\frac{3}{8}$	32.08	33.23	34.38	35.52	36.67	37.81	38.96	40.10	55.00
$1\frac{7}{16}$	33.54	34.74	35.94	37.14	38.33	39.53	40.73	41.93	57.50
$1\frac{1}{2}$	35.00	36.25	37.50	38.75	40.00	41.25	42.50	43.75	60.00
$1\frac{9}{16}$	36.46	37.76	39.06	40.36	41.67	42.97	44.27	45.57	62.50
$1\frac{5}{8}$	37.92	39.27	40.63	41.98	43.33	44.69	46.04	47.40	65.00
$1\frac{11}{16}$	39.38	40.78	42.19	43.59	45.00	46.41	47.81	49.22	67.50
$1\frac{3}{4}$	40.83	42.29	43.75	45.21	46.67	48.13	49.58	51.04	70.00
$1\frac{13}{16}$	42.29	43.80	45.31	46.82	48.33	49.84	51.35	52.86	72.50
$1\frac{7}{8}$	43.75	45.31	46.88	48.44	50.00	51.56	53.13	54.69	75.00
$1\frac{15}{16}$	45.21	46.82	48.44	50.05	51.67	53.28	54.90	56.51	77.50
2	46.67	48.33	50.00	51.67	53.33	55.00	56.67	58.33	80.00

WEIGHTS OF FLAT ROLLED IRON PER LINEAR FOOT.

(Continued)

Thickness in Inches.	9″	9¼″	9½″	9¾″	10″	10¼″	10½″	10¾″	12″
1/16	1.88	1.93	1.98	2.03	2.08	2.14	2.19	2.24	2.50
1/8	3.75	3.85	3.96	4.06	4.17	4.27	4.38	4.48	5.00
3/16	5.63	5.78	5.94	6.09	6.25	6.41	6.56	6.72	7.50
1/4	7.50	7.71	7.92	8.13	8.33	8.54	8.75	8.96	10.00
5/16	9.38	9.64	9.90	10.16	10.42	10.68	10.94	11.20	12.50
3/8	11.25	11.56	11.88	12.19	12.50	12.81	13.13	13.44	15.00
7/16	13.13	13.49	13.85	14.22	14.58	14.95	15.31	15.68	17.50
1/2	15.00	15.42	15.83	16.25	16.67	17.08	17.50	17.92	20.00
9/16	16.88	17.34	17.81	18.28	18.75	19.22	19.69	20.16	22.50
5/8	18.75	19.27	19.79	20.31	20.83	21.35	21.88	22.40	25.00
11/16	20.63	21.20	21.77	22.34	22.92	23.49	24.06	24.64	27.50
3/4	22.50	23.13	23.75	24.38	25.00	25.62	26.25	26.88	30.00
13/16	24.38	25.05	25.73	26.41	27.08	27.76	28.44	29.11	32.50
7/8	26.25	26.98	27.71	28.44	29.17	29.90	30.63	31.35	35.00
15/16	28.13	28.91	29.69	30.47	31.25	32.03	32.81	33.59	37.50
1	30.00	30.83	31.67	32.50	33.33	34.17	35.00	35.83	40.00
1 1/16	31.88	32.76	33.65	34.53	35.42	36.30	37.19	38.07	42.50
1 1/8	33.75	34.69	35.63	36.56	37.50	38.44	39.38	40.31	45.00
1 3/16	35.63	36.61	37.60	38.59	39.58	40.57	41.56	42.55	47.50
1 1/4	37.50	38.54	39.58	40.63	41.67	42.71	43.75	44.79	50.00
1 5/16	39.38	40.47	41.56	42.66	43.75	44.84	45.94	47.03	52.50
1 3/8	41.25	42.40	43.54	44.69	45.83	46.98	48.13	49.27	55.00
1 7/16	43.13	44.32	45.52	46.72	47.92	49.11	50.31	51.51	57.50
1 1/2	45.00	46.25	47.50	48.75	50.00	51.25	52.50	53.75	60.00
1 9/16	46.88	48.18	49.48	50.78	52.08	53.39	54.69	55.99	62.50
1 5/8	48.75	50.10	51.46	52.81	54.17	55.52	56.88	58.23	65.00
1 11/16	50.63	52.03	53.44	54.84	56.25	57.66	59.06	60.47	67.50
1 3/4	52.50	53.96	55.42	56.88	58.33	59.79	61.25	62.71	70.00
1 13/16	54.38	55.89	57.40	58.91	60.42	61.93	63.44	64.95	72.50
1 7/8	56.25	57.81	59.38	60.94	62.50	64.06	65.63	67.19	75.00
1 15/16	58.13	59.74	61.35	62.97	64.58	66.20	67.81	69.43	77.50
2	60.00	61.67	63.33	65.00	66.67	68.33	70.00	71.67	80.00

WEIGHTS OF FLAT ROLLED IRON PER LINEAR FOOT.

(Concluded)

Thickness in Inches.	11"	11¼"	11½"	11¾"	12"	12¼"	12½"	12¾"
1/16	2.29	2.34	2.40	2.45	2.50	2.55	2.60	2.66
1/8	4.58	4.69	4.79	4.90	5.00	5.10	5.21	5.31
3/16	6.88	7.03	7.19	7.34	7.50	7.66	7.81	7.97
1/4	9.17	9.38	9.58	9.79	10.00	10.21	10.42	10.63
5/16	11.46	11.72	11.98	12.24	12.50	12.76	13.02	13.28
3/8	13.75	14.06	14.38	14.69	15.00	15.31	15.63	15.94
7/16	16.04	16.41	16.77	17.14	17.50	17.86	18.23	18.59
1/2	18.33	18.75	19.17	19.58	20.00	20.42	20.83	21.25
9/16	20.63	21.09	21.56	22.03	22.50	22.97	23.44	23.91
5/8	22.92	23.44	23.96	24.48	25.00	25.52	26.04	26.56
11/16	25.21	25.78	26.35	26.93	27.50	28.07	28.65	29.22
3/4	27.50	28.13	28.75	29.38	30.00	30.63	31.25	31.88
13/16	29.79	30.47	31.15	31.82	32.50	33.18	33.85	34.53
7/8	32.08	32.81	33.54	34.27	35.00	35.73	36.46	37.19
15/16	34.38	35.16	35.94	36.72	37.50	38.28	39.06	39.84
1	36.67	37.50	38.33	39.17	40.00	40.83	41.67	42.50
1 1/16	38.96	39.84	40.73	41.61	42.50	43.39	44.27	45.16
1 1/8	41.25	42.19	43.13	44.06	45.00	45.94	46.88	47.81
1 3/16	43.54	44.53	45.52	46.51	47.50	48.49	49.48	50.47
1 1/4	45.83	46.88	47.92	48.96	50.00	51.04	52.08	53.13
1 5/16	48.13	49.22	50.31	51.41	52.50	53.59	54.69	55.78
1 3/8	50.42	51.56	52.71	53.85	55.00	56.15	57.29	58.44
1 7/16	52.71	53.91	55.10	56.30	57.50	58.70	59.90	61.09
1 1/2	55.00	56.25	57.50	58.75	60.00	61.25	62.50	63.75
1 9/16	57.29	58.59	59.90	61.20	62.50	63.80	65.10	66.41
1 5/8	59.58	60.94	62.29	63.65	65.00	66.35	67.71	69.06
1 11/16	61.88	63.28	64.69	66.09	67.50	68.91	70.31	71.72
1 3/4	64.17	65.63	67.08	68.54	70.00	71.46	72.92	74.38
1 13/16	66.46	67.97	69.48	70.99	72.50	74.01	75.52	77.03
1 7/8	68.75	70.31	71.88	73.44	75.00	76.56	78.13	79.69
1 15/16	71.04	72.66	74.27	75.89	77.50	79.11	80.73	82.34
2	73.33	75.00	76.67	78.33	80.00	81.67	83.33	85.00

The weights for 12" width are repeated on each page to facilitate making the additions necessary for plates wider than 12". Thus, to find the weight of 15¼" x ⅞", add the weights to be found in the same line for 3¼" x ⅞" and 12" x ⅞" = 9.48 + 35.00 = 44.48 lbs.

SQUARE AND ROUND IRON BARS.

Thickness or Diameter in Inches.	Weight of □ Bar One Foot long.	Weight of ○ Bar One Foot long.	Area of □ Bar in sq. inches.	Area of ○ Bar in sq. inches.	Circumference of ○ Bar in inches.
0					
$\frac{1}{16}$	.013	.010	.0039	.0031	.1963
$\frac{1}{8}$	.052	.041	.0156	.0123	.3927
$\frac{3}{16}$	.117	.092	.0352	.0276	.5890
$\frac{1}{4}$	.208	.164	.0625	.0491	.7854
$\frac{5}{16}$	.326	.256	.0977	.0767	.9817
$\frac{3}{8}$	.469	.368	.1406	.1104	1.1781
$\frac{7}{16}$	.638	.501	.1914	.1503	1.3744
$\frac{1}{2}$	.833	.654	.2500	.1963	1.5708
$\frac{9}{16}$	1.055	.828	.3164	.2485	1.7671
$\frac{5}{8}$	1.302	1.023	.3906	.3068	1.9635
$\frac{11}{16}$	1.576	1.237	.4727	.3712	2.1598
$\frac{3}{4}$	1.875	1.473	.5625	.4418	2.3562
$\frac{13}{16}$	2.201	1.728	.6602	.5185	2.5525
$\frac{7}{8}$	2.552	2.004	.7656	.6013	2.7489
$\frac{15}{16}$	2.930	2.301	.8789	.6903	2.9452
1	3.333	2.618	1.0000	.7854	3.1416
$\frac{1}{16}$	3.763	2.955	1.1289	.8866	3.3379
$\frac{1}{8}$	4.219	3.313	1.2656	.9940	3.5343
$\frac{3}{16}$	4.701	3.692	1.4102	1.1075	3.7306
$\frac{1}{4}$	5.208	4.091	1.5625	1.2272	3.9270
$\frac{5}{16}$	5.742	4.510	1.7227	1.3530	4.1233
$\frac{3}{8}$	6.302	4.950	1.8906	1.4849	4.3197
$\frac{7}{16}$	6.888	5.410	2.0664	1.6230	4.5160
$\frac{1}{2}$	7.500	5.890	2.2500	1.7671	4.7124
$\frac{9}{16}$	8.138	6.392	2.4414	1.9175	4.9087
$\frac{5}{8}$	8.802	6.913	2.6406	2.0739	5.1051
$\frac{11}{16}$	9.492	7.455	2.8477	2.2365	5.3014
$\frac{3}{4}$	10.21	8.018	3.0625	2.4053	5.4978
$\frac{13}{16}$	10.95	8.601	3.2852	2.5802	5.6941
$\frac{7}{8}$	11.72	9.204	3.5156	2.7612	5.8905
$\frac{15}{16}$	12.51	9.828	3.7539	2.9483	6.0868
2	13.33	10.47	4.0000	3.1416	6.2832
$\frac{1}{16}$	14.18	11.14	4.2539	3.3410	6.4795
$\frac{1}{8}$	15.05	11.82	4.5156	3.5466	6.6759
$\frac{3}{16}$	15.95	12.53	4.7852	3.7583	6.8722
$\frac{1}{4}$	16.88	13.25	5.0625	3.9761	7.0686
$\frac{5}{16}$	17.83	14.00	5.3477	4.2000	7.2649
$\frac{3}{8}$	18.80	14.77	5.6406	4.4301	7.4613
$\frac{7}{16}$	19.80	15.55	5.9414	4.6664	7.6576
$\frac{1}{2}$	20.83	16.36	6.2500	4.9087	7.8540
$\frac{9}{16}$	21.89	17.19	6.5664	5.1572	8.0503
$\frac{5}{8}$	22.97	18.04	6.8906	5.4119	8.2467
$\frac{11}{16}$	24.08	18.91	7.2227	5.6727	8.4430

SQUARE AND ROUND IRON BARS.

(Concluded)

Thickness or Diameter in Inches.	Weight of □ Bar One Foot long.	Weight of ○ Bar One Foot long.	Area of □ Bar in sq. inches.	Area of ○ Bar in sq. inches.	Circumference of ○ Bar in inches.
$\frac{3}{4}$	25.21	19.80	7.5625	5.9396	8.6394
$\frac{13}{16}$	26.37	20.71	7.9102	6.2126	8.8357
$\frac{7}{8}$	27.55	21.64	8.2656	6.4918	9.0321
$\frac{15}{16}$	28.76	22.59	8.6289	6.7771	9.2284
3	30.00	23.56	9.0000	7.0686	9.4248
$\frac{1}{16}$	31.26	24.55	9.3789	7.3662	9.6211
$\frac{1}{8}$	32.55	25.57	9.7656	7.6699	9.8175
$\frac{3}{16}$	33.87	26.60	10.160	7.9798	10.014
$\frac{1}{4}$	35.21	27.65	10.563	8.2958	10.210
$\frac{5}{16}$	36.58	28.73	10.973	8.6179	10.407
$\frac{3}{8}$	37.97	29.82	11.391	8.9462	10.603
$\frac{7}{16}$	39.39	30.94	11.816	9.2806	10.799
$\frac{1}{2}$	40.83	32.07	12.250	9.6211	10.996
$\frac{9}{16}$	42.30	33.23	12.691	9.9678	11.192
$\frac{5}{8}$	43.80	34.40	13.141	10.321	11.388
$\frac{11}{16}$	45.33	35.60	13.598	10.680	11.585
$\frac{3}{4}$	46.88	36.82	14.063	11.045	11.781
$\frac{13}{16}$	48.45	38.05	14.535	11.416	11.977
$\frac{7}{8}$	50.05	39.31	15.016	11.793	12.174
$\frac{15}{16}$	51.68	40.59	15.504	12.177	12.370
4	53.33	41.89	16.000	12.566	12.566
$\frac{1}{16}$	55.01	43.21	16.504	12.962	12.763
$\frac{1}{8}$	56.72	44.55	17.016	13.364	12.959
$\frac{3}{16}$	58.45	45.91	17.535	13.772	13.155
$\frac{1}{4}$	60.21	47.29	18.063	14.186	13.352
$\frac{5}{16}$	61.99	48.69	18.598	14.607	13.548
$\frac{3}{8}$	63.80	50.11	19.141	15.033	13.744
$\frac{7}{16}$	65.64	51.55	19.691	15.466	13.941
$\frac{1}{2}$	67.50	53.01	20.250	15.904	14.137
$\frac{9}{16}$	69.39	54.50	20.816	16.349	14.334
$\frac{5}{8}$	71.30	56.00	21.391	16.800	14.530
$\frac{11}{16}$	73.24	57.52	21.973	17.257	14.726
$\frac{3}{4}$	75.21	59.07	22.563	17.721	14.923
$\frac{13}{16}$	77.20	60.63	23.160	18.190	15.119
$\frac{7}{8}$	79.22	62.22	23.766	18.665	15.315
$\frac{15}{16}$	81.26	63.82	24.379	19.147	15.512
5	83.33	65.45	25.000	19.635	15.708

ANGLE IRON.

Weight Per Linear Foot.

6 x 6 x ⅝	24	Lbs.	2 x 2 x ¼	3½	Lbs.
5 x 5 x 9/16	16½	“	1¾ x 1¾ x 3/16	2¾	“
4 x 4 x ½	12½	“	1½ x 1½ x 3/16	2	‘
3½ x 3½ x 7/16	9	“	1¼ x 1¼ x 3/16	1½	“
3 x 3 x ⅜	7	“	1 x 1 x ⅛	1	“
2½ x 2½ x 5/16	5	“	¾ x ¾ x ⅛	⅝	“
2¼ x 2¼ x ¼	4¼	“			

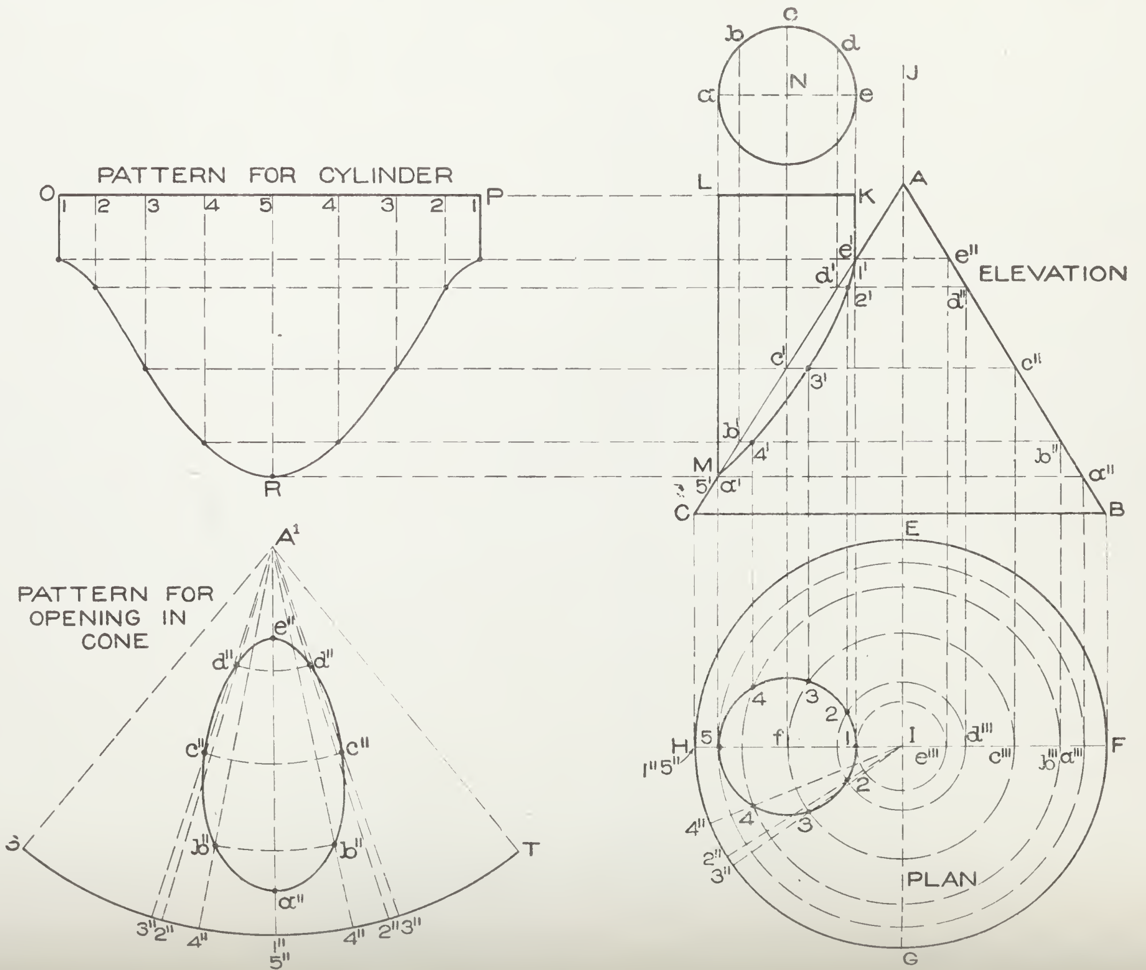
TEE IRON.

Weight Per Linear Foot.

5 x 8 x ⅝	30	Lbs.	2¼ x 2¼ x ¼	4	Lbs
7 x 6 x ⅝	30	“	2 x 2 x ¼	3¼	“
6 x 3 x ½	16½	“	1¾ x 1¾ x ¼	3	“
4 x 4 x ½	14	“	1½ x 1½ x ¼	2¾	“
3½ x 3½ x ½	12½	“	1¼ x 1¼ x ¼	2¼	“
3 x 3 x ⅜	7¾	“	1 x 1 x ⅛	1	“
2½ x 2½ x ½	8	“	¾ x ¾ x ⅛	⅜	“
2½ x 2½ x 5/16	5	“			

EXAMINATION PAPER

PLATE I


PATTERN FOR CYLINDER
O
P
1 2 3 4 5 4 3 2 1
R
ELEVATION
A
J
K
L
M
C
B
N
a b c d e
e' d' c' b' a'
1' 2' 3' 4' 5'
e'' d'' c'' b'' a''
PLAN
E
F
G
H
I
f
1 2 3 4 5
d''' c''' e''' b''a'''
1''5'' 2'' 3'' 4''
PATTERN FOR OPENING IN CONE
A¹
S
T
e'' d'' c'' b'' a''
3'' 2'' 4'' 1'' 5''

EXAMINATION PLATES.

PLATE I.

The plates of this Instruction Paper should be laid out the same size as explained in the course on Tinsmith's Work. Before starting these plates the student should first practice on other paper and make models of stiff cardboard to prove the accuracy of the patterns. When the problem is thoroughly mastered and understood, copy and send in your best drawing for examination and correction.

The first problem given is the intersection and development of a cylinder and a right cone, whose lines of axis run parallel to each other. First draw the base of the cone B C $4\frac{1}{4}$ inches long, placing B $1\frac{1}{2}$ inches from the border line, and the line C B $5\frac{1}{4}$ inches above the bottom line. Make the vertical height of the cone $3\frac{1}{2}$ inches, and draw the lines A B and A C. Through A draw the center vertical line J G, on which $2\frac{1}{2}$ inches below the base line C B establish the point I. Now with I as center and $\frac{1}{2}$ of C B as radius, describe the circle E F G H, which represents the plan of the cone. From A on the line A C measure down one inch as shown at e', from which erect a vertical line e' K $\frac{3}{4}$-inch high. From K draw the horizontal line K L equal to $1\frac{3}{8}$ inches, and from L drop a perpendicular intersecting the side of the cone at M. Directly above L K in its proper relative position $\frac{1}{4}$-inch above L K, with N as center draw the circle shown, which represents a section through L K. Through the center N draw the horizontal line a e. Now divide the half upper section N into an equal number of spaces as shown by a b c d e, from which points drop vertical lines intersecting the side of the cone A C at a' b' c' d' and e', and from these points, draw horizontal lines, intersecting the opposite side A B from a'' to e''.

The next step is to construct planes in plan as follows: From the various intersections a'' to e'' in elevation, drop vertical lines intersecting the horizontal line H F in plan drawn through the center I, at a''' b''' c''' d''' and e'''. Then using I as center and distances to points a''' to e''' as radii, strike the various circles shown. From the center of the section N in elevation extend the line intersecting H F in plan at f; then using f as center and N a

or N *e* in elevation as radius, describe the circle 1–3–5–3 in plan, cutting the various planes at 1, 2, 3, 4, and 5 on both sides. Now, from the various points of intersection on the various planes in plan, erect lines intersecting similarly lettered planes in elevation as shown at 1′, 2′, 3′, 4′, and 5′, through which trace a line as shown, which represents the line of intersection between the cylinder and right cone.

For the pattern for the cylinder proceed as follows: Extend the line K L of the cylinder in elevation as shown by O P, placing the distance 1 from the margin line 2½ inches. Now starting from 1, lay off on O P the stretchout of the section *f* in plan, the spaces being designated by similar figures on O P. From these small figures and at right angles to O P, draw lines which intersect with lines drawn from similarly numbered intersections on the line of intersection in elevation parallel to O P. Trace a line through points thus obtained. Then will O P R be the full pattern for the cylinder.

The pattern for the right cone is developed as has already been described in the Tinsmith's Course, and for that reason will be shown only the method how to obtain the pattern for the opening in the cone to miter with the cylinder. For this proceed as follows: Draw radial lines from I in plan through the intersections 1, 2, 3, 4, and 5, cutting the outer curve E F G H at 1″, 2″, 3″, 4″, and 5″, respectively. Now with A B in elevation as radius and A^1 as center, describe a short arc as S T. Place the arc as far above the margin line as the plan G, and have S T central between the plan and margin line. From A^1 drop a vertical line intersecting the arc S T at 1″. Now starting from 1″, set off on either side of the center line A^1 1″ the distances shown in plan from 1″ 5″ to 4″ to 2″ to 3″, as shown by similar numbers on S T. From these points draw radial lines to A^1. Now using A^1 as center and with radii equal to A *e*′, A *d*′, A *c*′, A *b*′, and A *a*′ in elevation, describe arcs intersecting respectively the radial lines A′ 1″ at *e*″, 2″ 2″ at *d*″ *d*″, 3″ 3″ at *c*″ *c*″, 4″ 4″ at *b*″ *b*″, and A′ 5″ at *a*″. Through these intersections trace a line as shown, which will be the desired opening.

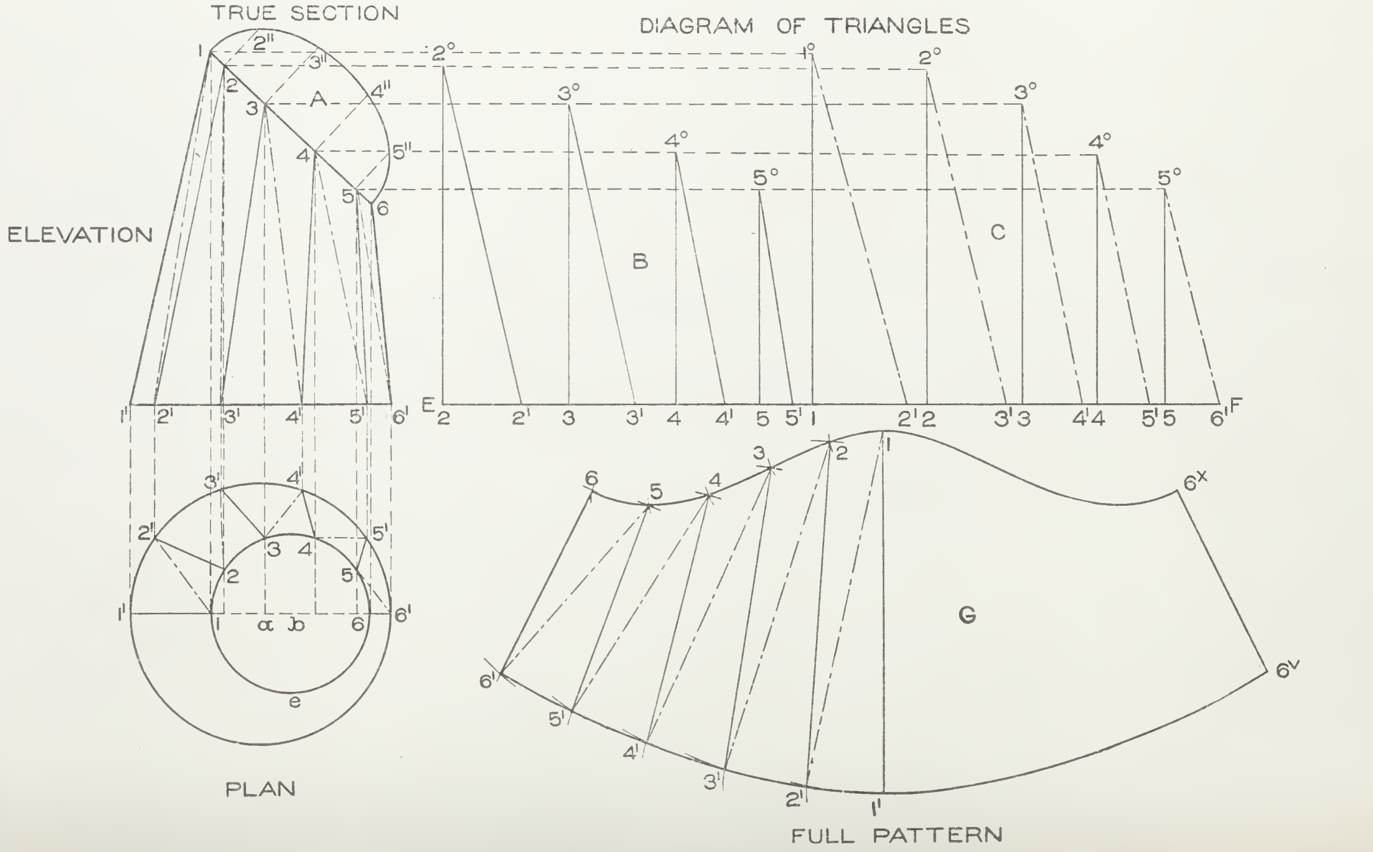
PLATE II.

This problem will give an examination on triangulation, being the development of an irregular solid whose bottom and top are round when viewed on horizontal planes, the top plane being cut off at an angle of 45° to the base line. First draw the base line 1′–6′, $2\frac{3}{4}$ inches long, placing the point 1′ 2 inches from the margin line, and the line 1′–6′ in the center of the sheet. Three inches above the bottom margin line draw the horizontal line 1′–6′ in plan; then with *a* as center and radius equal to one-half of 1′–6′ in elevation, draw the circle shown in its proper position below the line 1′–6′ in elevation. Now $\frac{1}{4}$ inch to the right of *a* on the line 1′–6′ in plan, establish *b*, which use as a center, and with $\frac{13}{16}$-inch radius describe the inner circle 1–3–6–*e*, which represents the upper horizontal plane of the irregular article. From the points 1 and 6 in plan, erect the vertical lines 1–1 and 6–6, making the height of the line 1–1 above the base line 1′–6′ in elevation $3\frac{3}{4}$ inches, and from the point 1 in elevation draw the line 1–6 at an angle of 45° intersecting the vertical line 6–6 at 6. Now draw lines from 1 to 1′ and from 6 to 6′, which completes the elevation.

Now divide the half plan into an equal number of parts (in this case 5) as shown by the small figures 1′ to 6′. In the same manner divide the inner circle into the same number of parts as shown from 1 to 6. Now draw solid lines from 1′ to 1, 2′ to 2, 3′ to 3, 4′ to 4, 5′ to 5, and 6′ to 6; and dotted lines from 1 to 2′, 2 to 3′, 3 to 4′, 4 to 5′, and 5 to 6′. From the intersections 1′ to 6′ on the outer plan, erect lines intersecting the base line in elevation from 1′ to 6′. In a similar manner, from the intersections on the inner curve in plan erect lines intersecting the top plane of the article from 1 to 6. Now connect lines in elevation from 2 to 2′, 3 to 3′, etc., although these lines are not necessary in the development of the pattern, but are given only to show their relationship to similar lines in plan. The solid and dotted lines in plan represent the bases of triangles, which will be constructed, whose altitudes are equal to similarly numbered vertical heights in elevation. The construction of these triangles is shown at B and C, B representing the triangles on solid lines in plan, and C the triangles on dotted lines in plan. Construct these triangles as follows: Extend the line 1′-6′ in elevation as E F, the point 6′ on E F to be placed 1 inch from the margin line. Now take the various distances of the dotted lines in plan as 6′ to 5, 5′ to 4 4′ to 3, 3′ to 2, and 2′ to 1, and place them on the line E F as shown by similar numbers. Now from the small figures 1, 2, 3, 4, and 5 on the line E F, erect lines, which intersect by lines drawn from similarly numbered intersections on 1-6 in elevation, parallel to E F, thus obtaining the points 1°, 2°, 3°, 4°, and 5°. Then draw lines from 1° to 2′, 2° to 3′, 3° to 4′, 4° to 5′, and 5° to 6′, which will represent the diagram of triangles on dotted lines in plan, the slant lines representing the true lengths on the finished article. In precisely the same manner obtain the diagram of triangles B on solid lines in plan. For example, take the distance of 2-2′ in plan, and place it as shown by 2–2′ on the line E F; from 2 at right angles to E F draw the line 2–2° equal to the vertical height to 2 in elevation, and draw a line from 2° to 2′, which is the true length on 2–2′ in plan. It now becomes necessary to obtain a true section on the line 1–6 in elevation. Therefore at right angles to 1–6 and from the various intersections 1 to 6, draw lines as shown. Now measuring in each instance from the line 1′–6′ in plan, take the various distances to points 2, 3, 4, and 5, and place them in A on similarly numbered lines, measuring in each instance from the line 1–6 in elevation, thus obtaining the points 2″ to 5″. A line traced as shown will be the half section on 1–6 in elevation.

For the pattern proceed as follows: Draw any line as 1–1′ in G equal to 1–1′ in elevation. Now with radius equal to 1′–2′ in plan, and 1′ in G as center, describe the arc 2′. Then using 1 in G as center, and 1°–2′ in C as radius, intersect the arc 2′ in G. Now with radius equal to 1–2″ in the true section, and 1 in G as center, describe the arc 2, which intersect by an arc struck from 2′ as center and with 2°–2′ in B as radius. Proceed in this manner, using alternately as radii, first the divisions in the outer curve in plan, then the hypothenuses or slant lines in C; the divisions in the true section A, then the length of the slant lines in B, following the numbers in regular order until the last line 6–6′ in G is obtained, this line being obtained from 6–6′ in elevation. Trace lines as shown from 1 to 6 and from 1′ to 6′, which give the half pattern. Trace the other half opposite the line 1–1′ as shown by $6^x 6^y$. Then will 6–1–6^x–6^y–1′–6′ be the full pattern with joint at 6–6′ in elevation

PLATE II


TRUE SECTION
DIAGRAM OF TRIANGLES
ELEVATION
A
B
C
E
F
PLAN
α
b
e
G
FULL PATTERN
1°
2°
3°
4°
5°
6x
6v

SHEET METAL WORK.

PART II.

PROBLEMS FOR LIGHT GAUGE METAL.

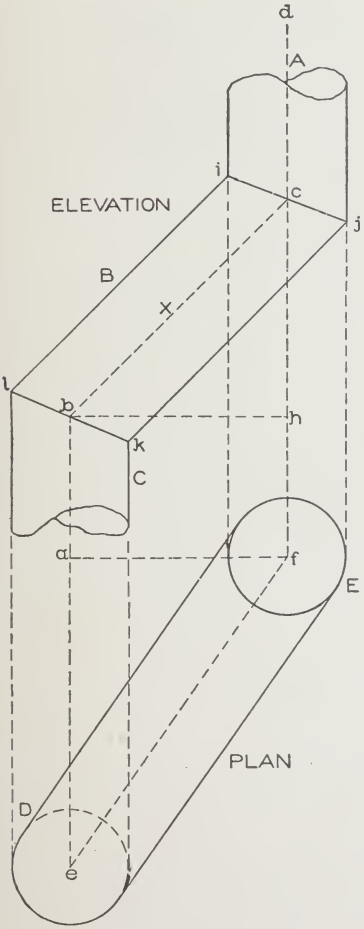


Fig. 65.

It is often the case that the sheet metal worker receives plans for vent, heat, or blower pipes to be constructed, in which the true lengths and angles are not shown but must be obtained from the plans or measurements at the building.

Figs. 65 and 66 show the principles employed for obtaining the true angles and lengths in oblique piping, it being immaterial whether the piping is round, square, or oval in section. The only safe way in obtaining these angles is to use the center line as a basis and after this line has been obtained, build the pipe around it, so to speak. In Fig. 65 let A B C represent the elevation of the elbow shown in plan by D E. Through the center of the pipes draw the center line *a b c d* which intersect the center lines of the pipe in plan at *e* and *f*. In elevation the rise of the middle piece B on the center line is equal to *h c* and projects to the right a distance equal to *b h*, shown in plan by *e f*; this same pipe projects forward in plan a distance equal to *e a*. While the miter lines in elevation *i j* and *k l* have been drawn straight, they would in reality show curved lines; those lines have not been projected as there is no necessity for doing so.

With the various heights and projections in plan and elevation the true length and true angles are obtained as shown in Fig.

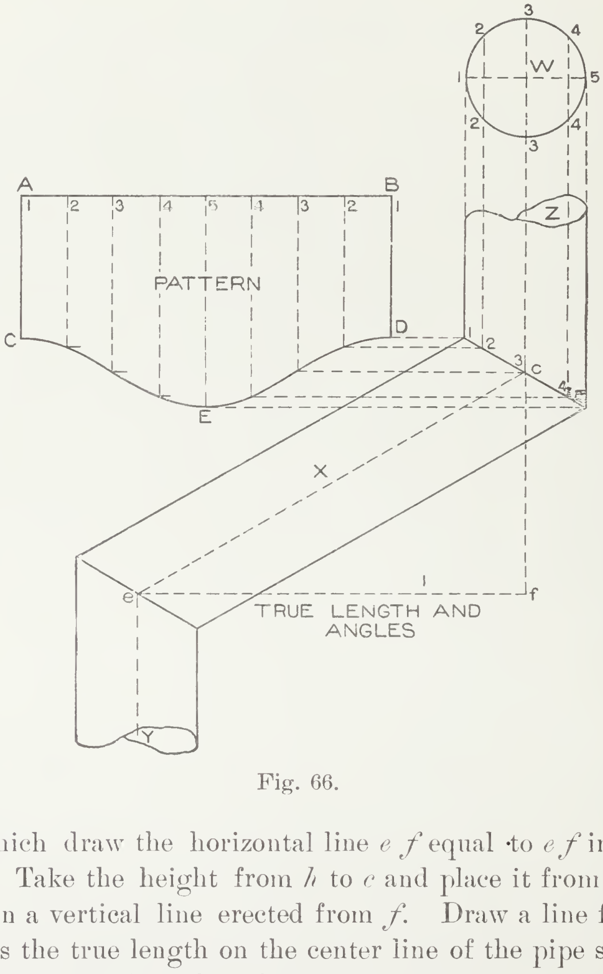


Fig. 66.

66, in which draw the horizontal line *e f* equal to *e f* in plan in Fig. 65. Take the height from *h* to *c* and place it from *f* to *c* in Fig. 66 on a vertical line erected from *f*. Draw a line from *e* to *c* which is the true length on the center line of the pipe shown by B in elevation in Fig. 65. From the points *c* and *e* in Fig. 66 draw perpendicular lines, making Y *e* X and X *c* Z = the true angles shown by *a b* X and X *c d* respectively in Fig. 65. On either side of the center line in Fig. 66 lay off the half diameter of the pipe as shown, and in its proper position draw the profile W.

Divide this into equal spaces and obtain the pattern A B D E C in the usual manner. As both angles are similar the miter cut C E D can be used for all of the patterns. In drawing this problem for practice make the diameter of the pipe 2 inches, the height from h to c $3\frac{3}{4}$ inches in Fig. 65, the projection b to h $3\frac{3}{4}$ inches, and the projection in plan e to a $5\frac{1}{4}$ inches.

Our next problem is that of a rain-water cut-off, a perspective view of which is shown in Fig. 67. While the miter cuts in this problem are similar to elbow work the intersection between the two beveled arms, and the cut-off or slide on the inside require attention. Make the diameter of the three openings each 2 inches; A to B (Fig. 68) $1\frac{1}{2}$ inches. From B at an angle of 45° draw B C $3\frac{1}{4}$ inches and C D 2 inches. From G draw the vertical miter line G h. Make the distance from B to T $\frac{1}{2}$ inch. Place the line d e of the cut-off $\frac{1}{8}$ inch above the line T U as indicated at a and the line e c to the right of h G, as indicated by b, a distance of $\frac{3}{16}$ inch. Parallel to G H draw c d giving slight play room between G H, intersecting e d and e c at d and c respectively. From c at right angles to d c, draw a line as shown, intersecting h G at f, which is the pivot on which the cut-off c d e will turn either right or left. The angles of the pipes on opposite sides are constructed in similar manner; A B C D E F G H I J K L M will be the elevation, N, the section on A M and O P R S the section on I J. B T U L shows how far the upper tube projects into the body under which the scoop e d c turns right and left to throw the rain water into either elbow as desired. The pattern for the upper piece A T U M is a straight piece of metal whose circumference is equal to N.

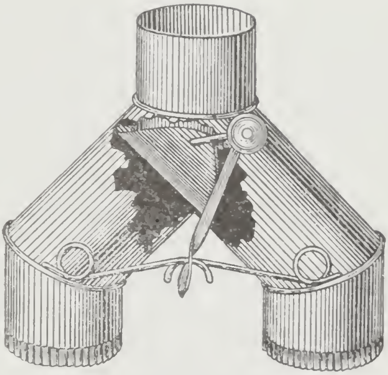


Fig. 67.

For the pattern for (A), divide the half section O P R into equal spaces as shown, from which erect lines intersecting the miter line H K as shown, and from which, parallel to K L and H G, draw lines intersecting the joint lines G h L as shown. As none of the lines just drawn intersect the corner *h*, it will be necessary to obtain this point on the half section O P R from which the stretchout of the pattern is taken. Therefore from *h*, parallel to L K draw *h h'* intersecting H K at *h'*, from which, parallel to K J, drop a line intersecting the profile O P R S at *h''*. At right angles to L K draw stretchout of O P R S as shown by similar numbers on T^1 U^1, through which at right angles to T^1 U^1 draw lines which are intersected by lines drawn at right angles to L K from similar intersections on G *h* L and H K. A line traced through points thus obtained as shown by X Y Z V W will be the pattern for (A). From *f* in the elevation at right angles to L K project a line intersecting the miter cut X Y Z at *f'* and *f''*. At *f'* and *f''* holes are to be punched in which the pivot *f* of the scoop *c d e* in elevation will turn.

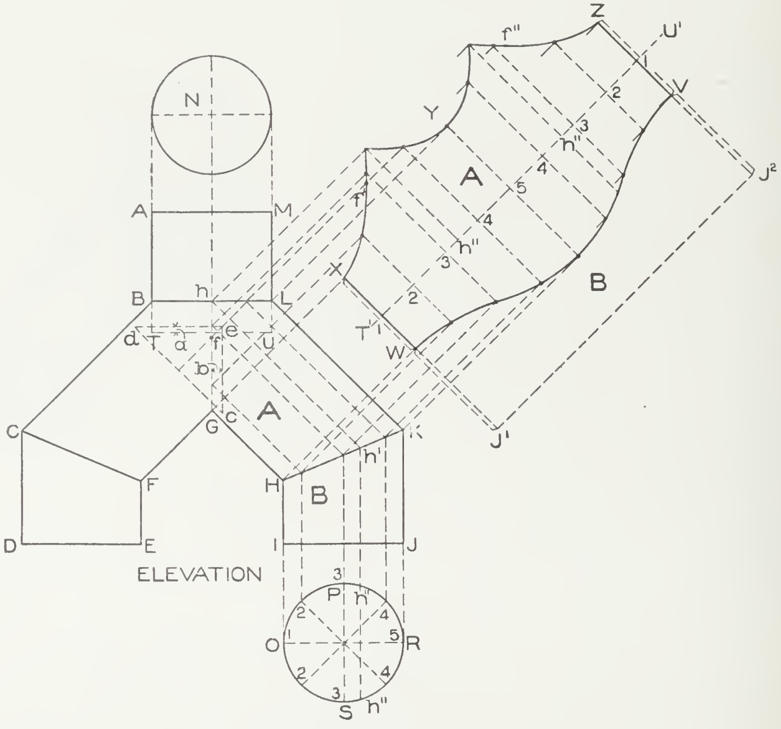


Fig. 68.

While the pattern for (B) can be obtained as that for (A) was obtained, a short method is to take the distance K to J and place it as shown from W to J^1 and V to J^2 on the lines of the pattern X W and Z V respectively extended. W V J^2 J^1 will be the pattern for B.

To avoid a confusion of lines in the development of the scoop or cut-off $c\ d\ e$, this has been shown in Fig. 69 in which $d\ e\ c$ is a reproduction of $d\ e\ c$ in Fig. 68. A true section of the scoop must now be drawn on $x\ e$ in Fig. 69 so that its dimensions will allow it to turn easily inside of the joint line G h in elevation in Fig. 68. Therefore draw any horizontal line as 4 5 in Fig. 69, at right angles to which from f draw a vertical line intersecting 4 5 at f'. Now take a distance $\frac{1}{16}$ inch less than one-half the diameter of O R in Fig. 68, and place it in Fig. 69 on either side of the line 4 5 on the vertical line just drawn as shown from f' to 2 and f' to 2′. Extend $d\ e$ till it intersects 4 5 at 4. Draw a line from 4 to 2′; by bisecting this line we obtain the line $a\ b$ intersecting 4 5 at i. Then with i as center and i 2′ as radius, describe the arc 2′ 2. From 2 and 2′ draw horizontal lines equal to $f'\ e$ as shown by 2 1 and 2′ 1′. Then will 1 4 1′ be the true section on $x\ e$. Divide the half section into equal spaces as shown from 1 to 4, from which erect lines intersecting $c\ e$ and $e\ d$. Extend $x\ e$ as $x\ j$, upon which place the stretchout of 1 4 1′ as shown by similar numbers on $x\ j$, through which draw vertical lines. These lines intersect with horizontal lines drawn from similar intersections on $d\ e\ c$. Through points thus obtained draw the line 1 n 1′ m which is the desired pattern. As the pivot hole f falls directly on line 2, then f''' f'' will be the position of the holes in the pattern. Laps must be allowed to all patterns.

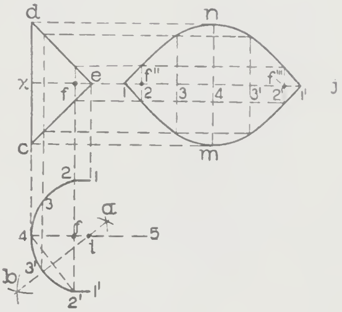


Fig. 69.

In putting up rectangular hot air pipe it is often the case that the pipe will be placed in the partition of one story, then has to fall forward and twist one quarter way around to enter the partition of the upper story which runs at right angles to the lower one. A perspective view showing this condition is shown in Fig. 70, where the upper opening turns one quarter on the lower one and leaning to the right as much as is shown in Fig. 71 in plan. This problem is known as a transition piece in a rectangular pipe.

Full size measurements are given in Fig. 71 which should be drawn one-half size. The height of the transition piece is 1 foot 8 inches, the size of the openings, each 4 × 10 inches turned as shown, two inches to the left and two inches above the lower section as shown. From the plan construct the front and side elevations as shown by the dotted lines. A B C D and E F G H will then be the front and side elevations of the transition piece respectively

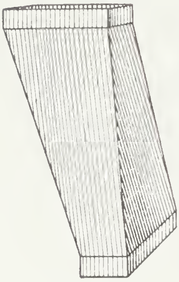


Fig. 70.

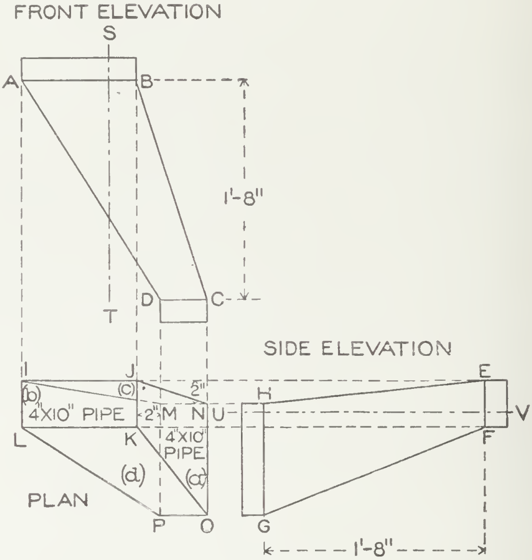


Fig. 71.

equal to 20 inches or 10 inches for practice. Number each side of the plan (*a*), (*b*), (*c*), and (*d*). Through the front and side elevations draw the vertical and horizontal lines S T and U V respectively at pleasure. These lines are only used as bases for measurements in determining the patterns. For the pattern for the side marked (a) in plan take the length of B C and place it on the vertical line B C in Fig. 72. Through the points B and C draw the horizontal lines E F and H G, making B F and B E, and C G and C H equal respectively to the distances measured from the line U V in Fig. 71 to points F, E, G, H. Draw lines from E to H and F to G in Fig. 72, which is the pattern for (a).

For the pattern for (b) in Fig. 71 take the distance of A D, and place it as shown by A D in Fig. 72; through A and D draw E F and H G, making A F and A E, and D G and D H equal respectively to the distances measured from the line U V in side elevation in Fig. 71 to points F, E, G, H. Draw lines from E to H and F to G in Fig. 72, which will be the pattern for (b). In similar manner obtain the patterns for (c) and (d) in plan in Fig. 71. The lengths of E H and F G are placed as shown by similar letters in Fig. 72, while the projections to A, B, C, D are obtained from A, B, C, D in front elevation in Fig. 71, measuring in each instance from S T.

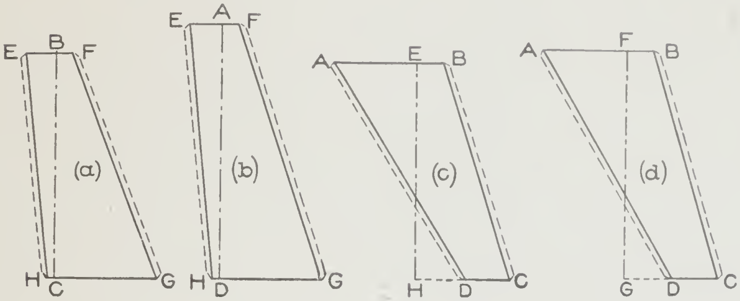


Fig. 72.

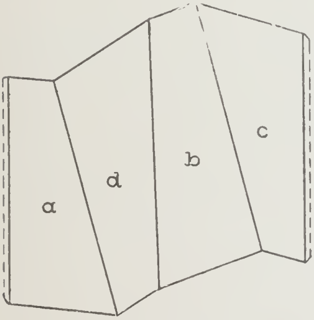


Fig. 73.

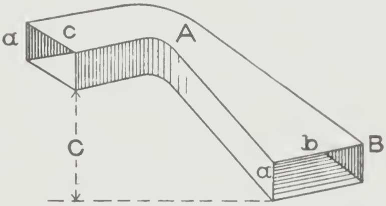


Fig. 74.

If desired the top and lower flange shown in the perspective in Fig. 70 can be added to the patterns in Fig. 72. Laps are allowed to the patterns to allow for double seaming at corners, if, however, the pattern should be required in one piece, it would only be necessary to join the various pieces in their proper positions as shown by *a d b c* in Fig. 73, which would bring the seam on the line J N in plan in Fig. 71.

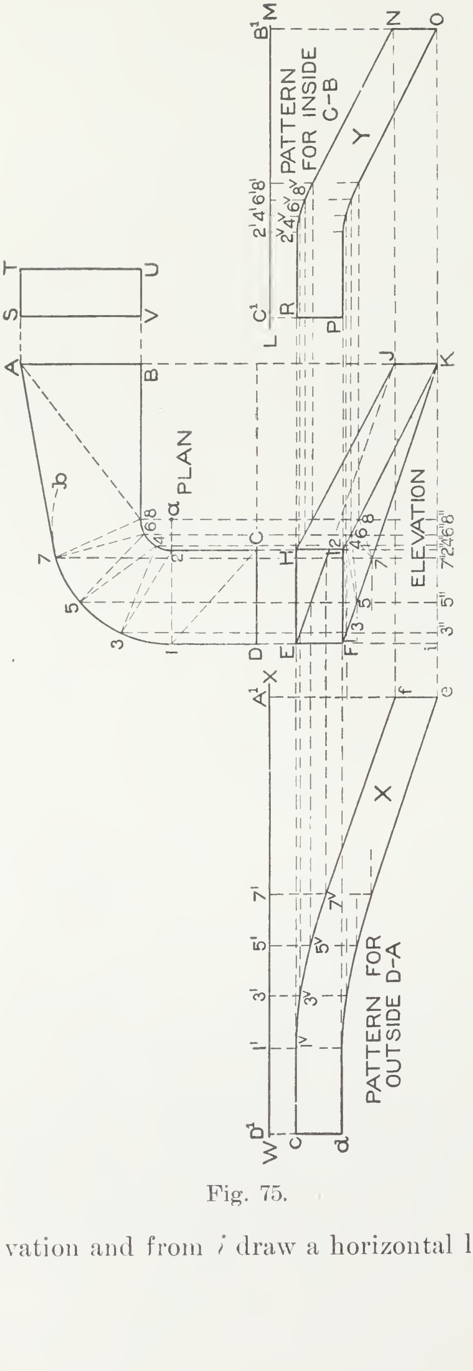


Fig. 75.

In Fig. 74 is shown a perspective view of a curved rectangular chute the construction of which arises in piping and blower work. The problem as here presented shows the sides *a* and *a* in vertical planes having the same height, while the bottom *b* has more width than the top *c*. The top opening is to rise above the bottom opening a given distance equal to C. First draw the plan and elevation as shown in Fig. 75, make A B equal to 2 inches, B 8 $2\frac{1}{2}$ inches; with a radius equal to $\frac{1}{2}$ inch, with *a* as center draw the quarter circle 8 2. From 2 draw the vertical line 2 C equal to $1\frac{3}{8}$ inches and draw C D equal to $1\frac{1}{2}$ inches. Make D 1 equal to C 2 and using *a* as center and *a* 1 as radius draw the arc 1 *b*. From A draw a line tangent to 1 *b* as A 7. A B C D will be the plan of the chute. In line with A B draw the section S T U V. In line with D C draw the section E F I H as shown. Place the desired rise of the chute as shown by F *i* in elevation and from *i* draw a horizontal line as *i* K, which intersect by a line drawn from A B in plan as shown. Make K J equal to F E and draw the lines F K, K I, and E J, J H. F E J K is the elevation of the outside curve, H I K J the inside curve, F I K the bottom, and E H J the top.

Having the plan and elevation in position we will first draw the pattern for the two vertical sides. For the pattern for the side of the chute shown by B C in plan proceed as follows: Divide the inner curve 2 to 8 into equal parts as shown by 2-4-6 and 8, from which points drop lines intersecting the inside of the chute in plan H J K I as shown. At right angles to J K draw L M, upon which place the stretchout of B C in plan as shown by similar letters and numbers on L M, through which draw vertical lines which intersect lines drawn parallel to L M from H J. Through points thus obtained draw the line R 2^V 4^V 6^V 8^VN. The same method can be employed for the curve P O, but as the height H I and J K are equal, having a common profile B C, take the height of H I or J K and place it on vertical lines as R P and N O and trace the curve R N as shown by P O. N O P R is the pattern for C B in plan; To obtain the pattern for the outside curve divide the curve 1-7 into equal parts as shown, from which drop vertical lines intersecting similar points in E J K F, in elevation at right angles to E F draw W X, upon which place the stretchout of D A in plan as shown. From the divisions on W X drop vertical lines, which intersect by lines drawn from similar numbered intersections on E J. Trace a line through these points as shown by *e f* and draw *d e* as explained in connection with the inside pattern. *c d e f* is the pattern for the outside of the chute shown in plan by D A.

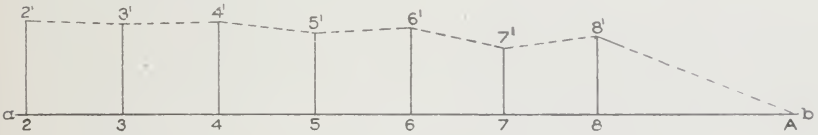


Fig. 76.

As both the top and bottom of the chute have the same bevel, the pattern for one will answer for the other. Connect opposite points in plan as shown from C to 1 to 2 to 3 up to 8, then to A. In similar manner connect similar points on the bottom in elevation as shown from 1 to 2 up to K. The lines in plan represent the bases of the sections whose altitudes are equal to the various heights in elevation, measured from *i* K. Take the various lengths from 2 to 3 to 4 to 5 to 6 to 7 to 8 to A in plan and place them as shown by similar numbers on the horizontal line *a b* (Fig. 76); through *a b* draw vertical lines, equal in height to similar numbers in elevation, in Fig. 75, measured from the line *i* K. For example take the distance 4 5 in plan and place it as shown by 4 5 in Fig. 76. Erect perpendiculars 4 4′ and 5 5′ equal to 4″ 4 and 5″ 5 in elevation in Fig. 75. Draw a line from 4′ to 5′ in Fig. 76, which is the true length of 4 5 in plan in Fig. 75. Proceed in similar manner for the balance of the sections. Take a tracing of 1 2 C D in plan and place it as shown by 1, 2, C, D in Fig. 77. Now using 1 as

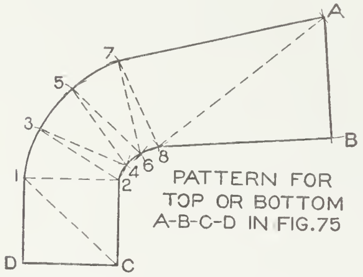


Fig. 77.

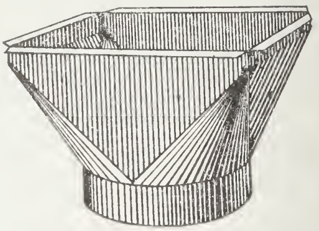


Fig. 78.

center and 1^{V} 3^{V} in (x), in Fig. 75, as radius, describe the arc at 3, in Fig. 77, which is intersected by an arc, struck from 2 as center, and 2′ 3′, in Fig. 76, as radius. Now with radius equal to 2^{V} 4^{V} in (Y) in Fig. 75 and 2 in Fig. 77 as center, describe the arc at 4 which is intersected by an arc, struck from 3 as center and 3′ 4′, Fig. 76, as radius. Proceed in this manner, using alternately as radius, first the divisions in the pattern (X), Fig. 75, then the slant lines in Fig. 76, the divisions in the pattern (Y), Fig. 75, then again the lines in Fig. 76 until the line 7 8, Fig. 77, has been obtained. Then using 7 as center, with a line equal to $7^{V}f$ in (X), Fig. 75, as radius, describe the arc A, Fig. 77, which is intersected by an arc struck from 8 as center and 8′ A, Fig. 76, as radius. Then with radius, equal to 8^{V} N in (Y), Fig. 75, and 8, Fig. 77, as center, describe the arc B, which is intersected by an arc, struck from A as center and A B in plan in Fig. 75 as radius. Trace lines through points thus obtained in Fig. 77, and A B C D will be the desired pattern. Laps must be allowed on all patterns for double seaming the corners.

In Fig. 78 is shown a perspective view of a hopper register box usually made from bright tin or galvanized iron in hot air piping. In drawing this problem, the student should first draw the half plan, making the semicircle $3\frac{3}{4}$ inches diameter, and placing it directly in the center of the rectangular top, which is $3\frac{3}{4}$ inches wide and $5\frac{3}{4}$ inches long. Draw the elevation from the plan as shown by A B C D E F G H, making the vertical height V W, $2\frac{1}{4}$ inches, and the flanges at the top and bottom each $\frac{1}{2}$ inch. I K L M in plan is the horizontal section on A B in elevation and O P R the section on E F.

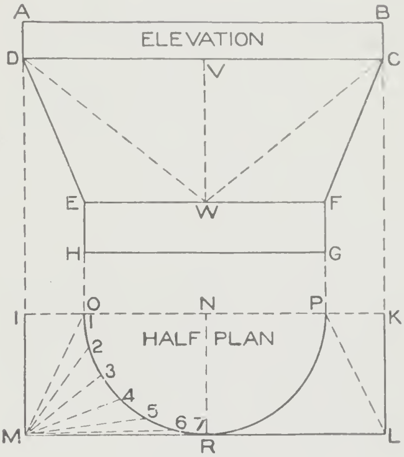


Fig. 79.

The pattern will be developed by triangulation, and the first step is to develop a set of triangles. Divide the quarter circle O R into equal spaces, as shown by the numbers 1 to 7 in plan, from which draw lines to the apex M. These lines represent the bases of triangles whose vertical height is equal to V W in elevation. Therefore, in Fig. 80, draw any horizontal line as T U, upon which place the various lengths M 1, M 2, M 3, etc.) Fig. 79) as shown by similar numbers on T U. From T U erect the line T S equal to the vertical height V W (Fig. 79). Then draw the hypotenuses S 1, S 2, S 3, etc., in Fig. 80, which represent the true lengths of similar numbered lines in plan in Fig. 79. For the half pattern with seams on I O and P K in plan, take a tracing of D V W in elevation and place it as shown by D V 7 in Fig. 81. Now using D as center, and with radii equal to the various slant lines in Fig. 80 from S 1 to S 7 strike small arcs as shown from 1 to 7 in Fig. 81. Set the dividers

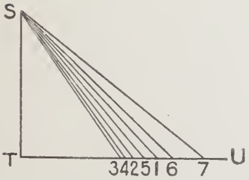


Fig. 80.

equal to the spaces contained in O R, in Fig. 79, and starting from point 7, in Fig. 81, step from one arc to another until 1 is obtained. Then using 1 as center and E D (Fig. 79) as radius describe the arc D′ in Fig. 81. With D as center and M I in plan in Fig. 79 as radius, draw another arc intersecting the one previously drawn at D′. Draw a line from 1 to D′ to D in Fig. 81, 7 1 D′ D V is the quarter pattern, and the left-hand side of the figure may be made by tracing the quarter pattern reversed as shown by V C D″ 1′ 7. Take the distance of the flange D A in elevation in Fig. 79 and place it at right angles to the line D′ D, D C, C D″ as shown respectively by A″ A′, A A^O and A^V A^X, which completes the half pattern with laps allowed as shown

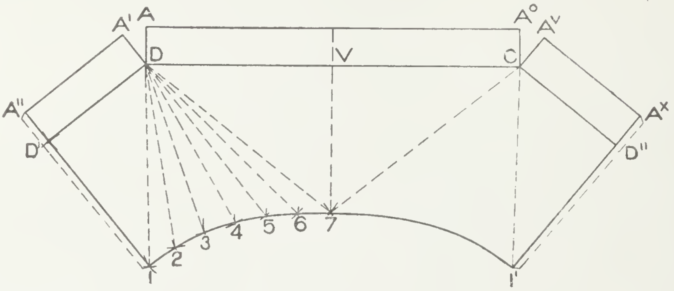


Fig. 81.

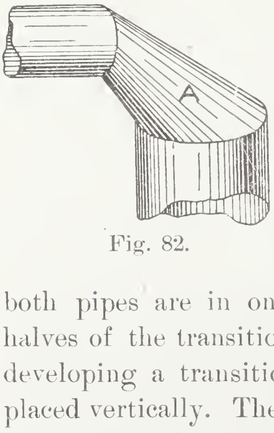


Fig. 82.

The pattern for the collar E F G H in elevation in Fig. 79 is simply a straight strip of metal, equal to the circumference of O P R in plan.

It is often the case that two unequal pipes are to be connected by means of a transition piece as shown by A in Fig. 82, the ends of the pipes being cut at right angles to each other. As the centers of both pipes are in one line when viewed in plan, making both halves of the transition piece equal, the problem then consists of developing a transition piece, from a round base to a round top placed vertically. Therefore in Fig. 83 draw 1 5 equal to $2\frac{1}{4}$ inches, and at an angle of 45° draw 5 6 $1\frac{3}{4}$ inches. At right angles to 1 5 draw 6 10 4 inches long and draw a line from 10 to 1. On 1 5 draw the semicircle 1 3′ 5, and on 6 10 draw the semicircle 6 8′ 10.

Divide both of these into equal spaces as shown, from which draw lines perpendicular to their respective base lines. Connect opposite points as shown by the dotted lines, and construct a diagram of

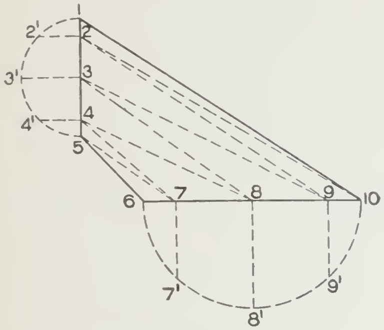


Fig. 83.

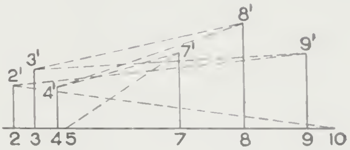


Fig. 84.

sections as shown in Fig. 84 whose bases and heights are equal to similar numbered bases and heights in Fig. 83. For example, take the distance 4 8 and place it as shown by 4 8 in Fig. 84, from which points erect the vertical lines 4 4′ and 8 8′ equal to 4 4′ and 8 8′ in Fig. 83. Draw a line from 4′ to 8′, Fig. 84, which is the true

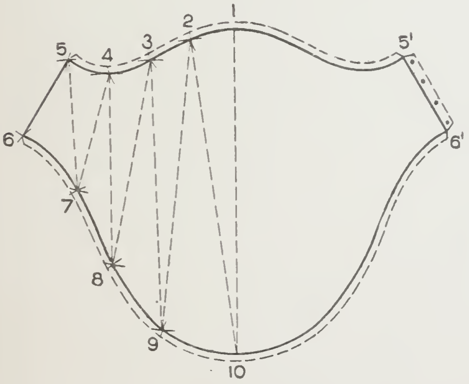


Fig. 85.

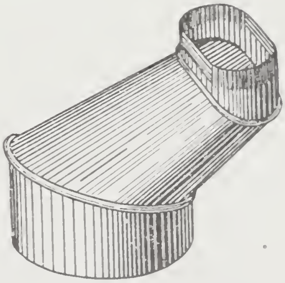


Fig. 86.

length on similar line in Fig. 83. For the pattern take the distance of 1 10 and place it as shown by 1 10 in Fig. 85. Using 1 as center, and 1 2′, Fig. 83, as radius, describe the arc 2 in Fig. 85; intersect it by an arc struck from 10 as center and 10 2′, Fig. 84, as radius. Then using 10 9′ in Fig. 83 as radius, and 10, Fig. 85, as center, describe the arc 9, and intersect it by an arc struck from 2 as center, and 2′ 9′, Fig. 84, as radius. Proceed in this manner using alternately as radii, first the divisions in the half profile 1 3′ 5, Fig. 83, then the length of the proper hypotenuse in Fig. 84, then the divisions in 6 8′ 10 in Fig. 83; then again the hypotenuse in Fig. 84 until the line 5 6 in Fig. 85 has been obtained, which is equal to 5 6 in Fig. 83. Laps should be allowed for riveting and seaming as shown.

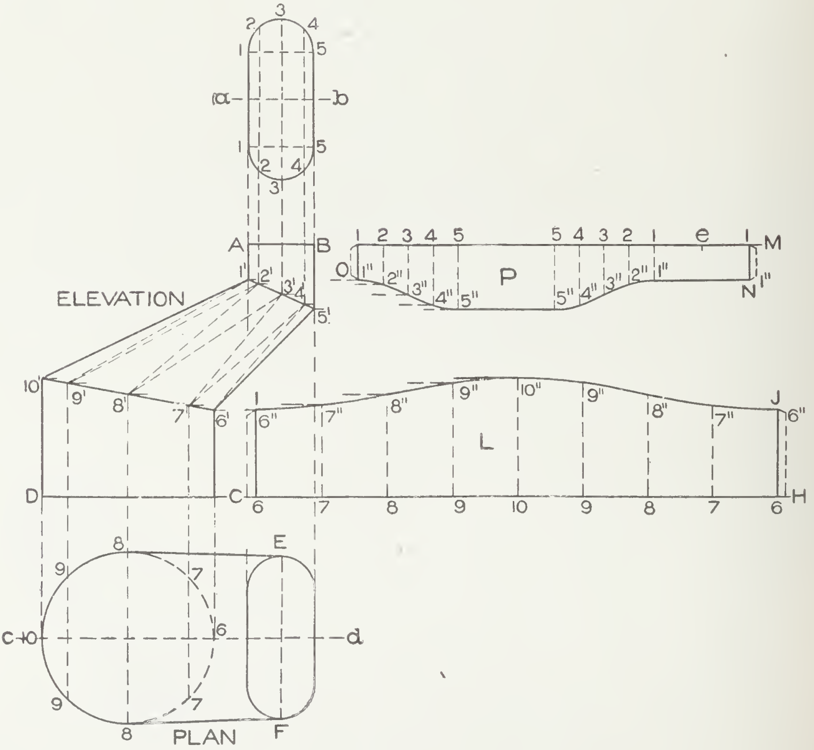


Fig. 87.

In Fig. 86 is shown a perspective of an offset connecting a round pipe with an oblong pipe, having rounded corners.

The first step is to properly draw the elevation and plan as shown in Fig. 87. Draw the horizontal line A B equal to one inch, B 5′ one inch, and from 5′ at an angle of 45° draw 5′ 6′ equal to $2\frac{1}{4}$ inches and 6′ C $1\frac{1}{4}$ inches. Make the diameter C D $2\frac{3}{4}$ inches and D 10′ $1\frac{3}{4}$ inches. Make A 1′ $\frac{1}{2}$ inch and draw a line from 1′ to

10′ which completes the elevation. Directly above the line A B draw the section of the oblong pipe, making the sides 1 1 and 5 5 equal to 1½ inches, to which describe the semicircles on each end as shown. In similar manner draw the section on D C, which is shown by 6 8 10 8. A duplicate of the oblong pipe is also shown in plan by E F, showing that the centers of the pipe come in one line, making both halves symmetrical.

The patterns for the pipes will first be obtained. Divide the semicircular ends of the oblong section into equal parts, in this case four, also each of the semicircles of the round pipe in similar number of parts as shown respectively from 1 to 5 and 6 to 10. Draw vertical lines from these intersections cutting the miter line of the oblong pipe at 1′ 2′ 3′ 4′ 5′ and the miter line of the round pipe at 6′ 7′ 8′ 9′ and 10′. In line with A B draw B M, upon which place the stretchout of the oblong pipe as shown by similar numbers; from B M drop vertical lines intersecting the lines drawn parallel to B M from similarly numbered points on 1′ 5′. Trace a line through points thus obtained, and P N O will be the pattern for the oblong pipe. Now take the stretchout of the round pipe, and place it on C H; erect vertical lines as shown intersecting the lines drawn parallel to C H from similar intersections on 6′ 10′. I J H C is the pattern for the round pipe.

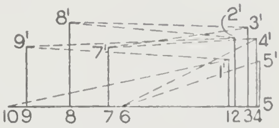


Fig. 88.

The transition piece 1′ 5′ 6′ 10′ will be developed by triangulation, and it is usual to obtain true sections on the lines 1′ 5′ and 6′ 10′; however, in this case it can be omitted because we have the true lengths of the various divisions on the lines 1′ 5′ and 6′ 10′ in the miter cuts in P and L respectively.

The next step is to obtain a diagram of sections giving the true lengths, for which proceed as follows: Connect opposite points in elevation as shown from 1′ to 9′ to 2′ to 8′ to 3′ etc., as shown. For example draw center lines through the oblong and round sections as shown by *a b* and *c d* respectively, and take the length of 1′ 10′ in elevation and place it as shown from 1 to 10 in Fig. 88. From 1 draw the vertical line 1 1′ equal to the height of 1 in the oblong section in Fig. 87 above the center line *a b*. As point 10 in plan has no height, it falls on the center line *c d* in plan, then draw a line from 1′ to 10 in Fig. 88. Now take the distance from 1′ to 9′ in elevation, Fig. 87, and place it as shown from 1 to 9 in Fig. 88. Erect the lines 1 1′ and 9 9′ equal to points 1 and 9 in the oblong and round sections in Fig. 87, measured respectively from the lines *a b* and *c d*. Draw a line from 1′ to 9′ in Fig. 87. Proceed in this manner until all of the sections are obtained. For the pattern proceed as shown in Fig. 89, in which draw any vertical line as *e* 10 equal to 1′ 10′ in elevation in Fig. 87. Now, with one-half of 1 1 in pattern P as *e* 1 as radius, and *e* in Fig. 89 as center, describe the arc 1 which is intersected by an arc struck from 10 as center and 10 1′, in Fig. 88 as radius. With radius equal to 10″ 9″ in pattern L in Fig. 87, and 10 in Fig. 89 as center describe the arc 9, which is intersected by an arc struck from 1 as center and 1′ 9′, in Fig. 88 as radius. Now, using as radius 1″ 2″ in pattern P in Fig. 87 and 1 in Fig. 89 as center, describe the arc 2 which is intersected by an arc struck from 9 as center and 9′ 2′ in Fig. 88 as radius.

Proceed in this manner, using alternately as radii, first the divisions in the pattern cut I J, Fig. 87, then the length of the slant lines in Fig. 88, the divisions in the cut O N in Fig. 87, then again the slant lines in Fig. 88 until the line 5 6 in pattern, Fig. 89, has been obtained. Then using 5 as center and 1 *e* in P, Fig. 87, as radius, describe the arc *e′* in Fig. 89, and intersect it by an arc struck from 6 as center and 6′ 5′ in elevation in Fig. 87 as radius. Draw lines through the various intersections in Fig. 89; 10 *e e′* 6 is the half pattern. By tracing it opposite the line *e* 10, as shown by *e* 1′ 5′ *e″* 6′ 10, the whole pattern, *e′ e e″* 6′ 10 6, is found. Laps should be allowed on all patterns for seaming or riveting both in Figs. 87 and 89.

In Fig. 90 is shown a perspective view of a three-way branch round to round, the inlet A being a true circle, and the outlets B, C, and D also being true circles, the centers of which are in the same vertical plane, thus making both sides of the branch symmetrical.

First draw the elevation and the various sections as shown in Fig. 91. Draw the center line *a b*. From *b* draw the center line of the branch C at an angle of 58° as shown by *b d*. Make the center lines *a b* and *b d* each $3\frac{1}{2}$ inches long. Make the half diameter of the branch B at the outlet $\frac{3}{4}$ inch, and the full diameter of the branch C at the outlet $1\frac{1}{2}$ inches placed on either side of and at right angles to the center lines. Draw a line from *e* to *f*, and with *i* and *h* as centers and radii equal to $\frac{3}{4}$ inch draw arcs intersecting each other at *c*. Draw lines from *i* to *c* to *h*. In similar manner obtain A and the opposite half of B. A B C is the elevation of the three branches whose sections on outlet lines are shown respectively by G F and E and whose section on the inlet line is shown by D.

The next step is to obtain a true section on the miter line or line of joint *b c*. Knowing the height *b c* and the width at the bottom, which is equal to the diameter of D, the shape can be drawn at pleasure as shown in Fig. 92, *b c* is drawn equal to *b c*, Fig. 91, while *b d* and *b a* are equal to the half diameter D in Fig. 91. Now through *a c d* in Fig. 92 draw the profile at pleasure as shown, which represents the true section on *c b* in Fig. 91.

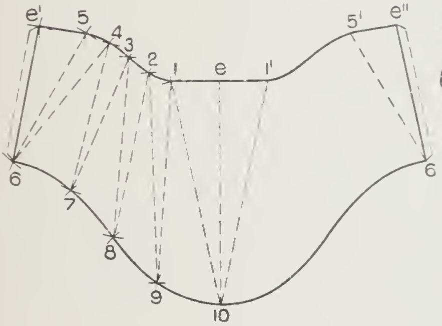


Fig. 89.

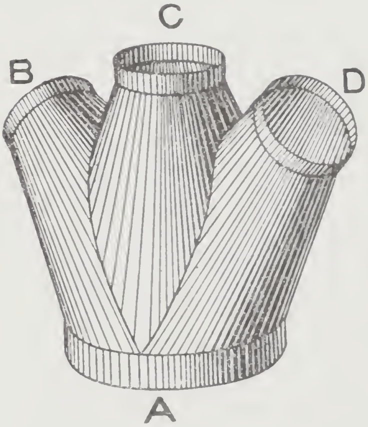


Fig. 90.

As the side branches A and C are alike, only one pattern will be required, also a separate pattern for the center branch both of which will be developed by triangulation. To obtain the measurements for the sections for the center branch B, proceed as shown in Fig. 93 where 1 4 5 8 is a reproduction of one-half the branch B in Fig. 91. As the four quarters of this center branch are alike only one quarter pattern will be developed; then, if desired, the quarter patterns can be joined together, forming one pattern. Now take a tracing of *c b a*, Fig. 92, and place it on the line 5 8 as shown in Fig. 93. Similarly take a tracing of the quarter profile F in Fig. 91 and place it on the line 4 1 in Fig. 93. Divide the two profiles 1′ 4 and 5 8′ each into the same number of spaces as shown respectively by points 1′ 2′ 3′ 4 and 5 6′ 7′ 8′, from which points at right angles to their respective base lines 1 4 and 5 8 draw lines intersecting the base lines at 1 2 3 4 and 5 6 7 8. Now draw solid lines from 3 to 6 and 2 to 7 and dotted lines from 3 to 5, 2 to 6, and 1 to 7. These solid and dotted lines represent the bases of the sections whose altitudes are equal to the various heights of the profiles in Fig. 93. The slant lines in Fig. 94 represent the true distances on similar lines in Fig. 93, as those in Fig. 95 represent the true distances on dotted lines in Fig. 93.

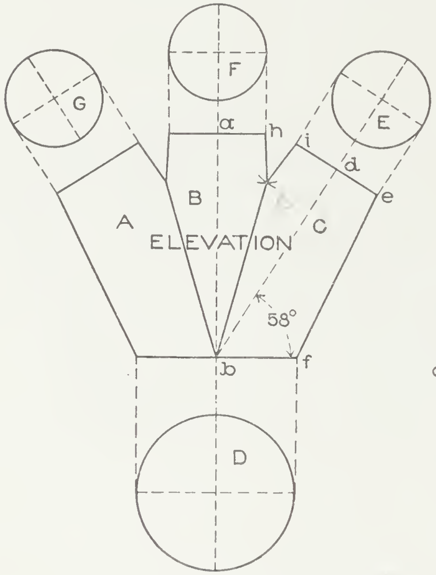


Fig. 91.

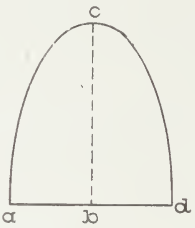


Fig. 92.

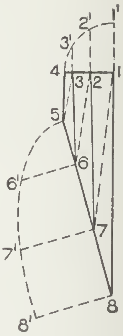


Fig. 93.

For the pattern take the length of 1′ 8′, Fig. 94, and place it as shown by 1 8 in Fig. 96, and using 8 as center and 8′ 7′ in Fig. 93 as radius draw the arc 7, which intersect by an arc struck from 1 as center and 1′ 7′ in Fig. 95 as radius. Then using 1′ 2′ in Fig. 93 as radius draw the arc 2, which intersect by an arc struck from 7 as center and 7′ 2′ in Fig. 94 as radius. Proceed in this manner until the line 4 5 in Fig. 96 has been obtained which equals 4 5 in Fig. 93. Trace a line through points thus obtained in Fig. 96, then will 1 4 5 8 1 give the quarter pattern.

If the pattern is desired in one piece trace as shown by similar figures, to which laps must be allowed for riveting.

As the two branches A and C in Fig. 91 are alike, one pattern will answer for the two. Therefore let 1 7 8 11 14 in Fig. 97 be a reproduction of the branch C in Fig. 91. Now take a tracing of *a b c* in Fig. 92 and place it as shown by 11′ 11 8 in Fig. 97; also take a tracing of the half section E and the quarter section D in Fig. 91 and place them as shown respectively by 1 4′ 7 and 11 11′ 14 in Fig. 97. Now divide the two lower profiles 8 11 and 11′ 14 each into 3 equal parts, and the upper profile 7 4′ 1 into 6 equal parts as shown by the small figures 8 to 11′, 11′ to 14 and 1 to 7. From these points, at right angles to the various base lines, draw lines, intersecting the base lines as shown by similar numbers. Draw solid and dotted lines as shown, and construct the sections on solid lines as shown in Fig. 98 and the sections on dotted lines as shown in Fig. 99 in precisely the same manner as described in connection with Figs. 94 and 95.

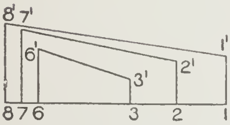


Fig. 94.

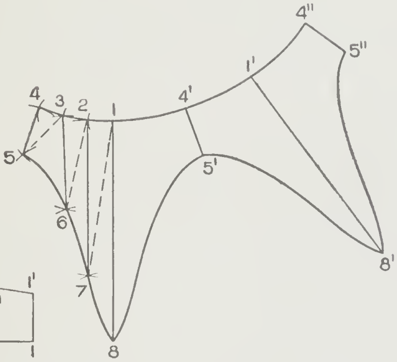


Fig. 96.

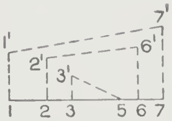


Fig. 95.

In Fig. 100 is shown the pattern shape (to which laps must be allowed for riveting) obtained as was the development of Fig. 96. First draw the vertical line 1 14, Fig. 100, equal to 1 14 in Fig. 97. Then use alternately as radii, first the divisions in 1 4′ 7 in Fig. 97, the proper slant line in Figs. 98 and 99 and the divisions in 11′ 14 until the line 4 11, Fig. 100, is obtained. Starting from the point 11 use as radii in their regular order the distances marked off between 11′ and 8, Fig. 97, then the proper slant lines in Figs. 98 and 99, the distances shown in the semicircle, 1 4′ 7, Fig. 97, until the line 7 8, Fig. 100, is drawn equal to 7 8, in Fig. 97. Then

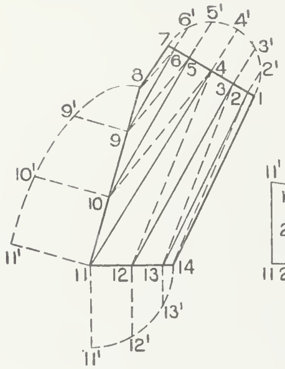


Fig. 97.

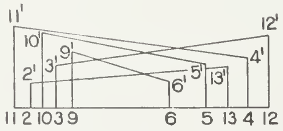


Fig. 98.

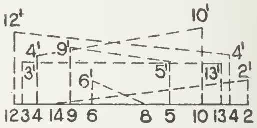


Fig. 99.

1 7 8 11 14, Fig. 100, will be the half pattern. If the pattern is desired in one piece trace 1 7′ 8′ 11′ 14 opposite the line 1 14 as shown.

In Fig. 101 is shown a perspective view of a two-branch fork oval to round, commonly used as breeching for two boilers. As

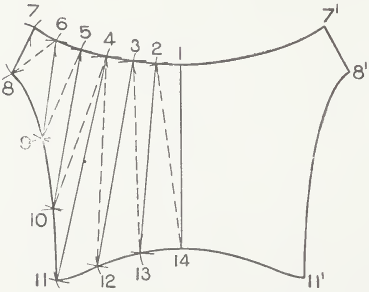


Fig. 100.

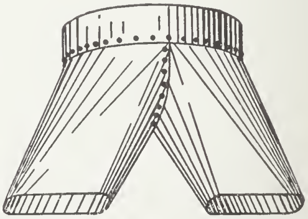


Fig. 101.

both halves of the fork are symmetrical the pattern for one will answer for the other.

While the side elevation shown in Fig. 102 is drawn complete, it is only necessary in practice, to draw one half as follows, and then, if desired, the other half elevation can be traced opposite to the center line E J. First draw J B, $1\frac{1}{2}$ inches, equal to the half diameter of the outlet, and the vertical center height J V, $2\frac{1}{4}$ inches. Establish the height of the joint J E one inch, and the desired projection V D on the base line $1\frac{1}{4}$ inches. Draw the length of the inlet D C $2\frac{3}{4}$ inches, and draw a line from C to B and D to E. Draw a similar figure opposite the line J E, and A B C D E F G shows the side elevation of the fork. In their proper position below A B draw the sections M and N whose semicircular ends are struck from *a b c* and *d* with radii equal to $\frac{1}{2}$ inch. Now draw an end elevation in which the true section on J E is obtained. Draw the center line *f e* and extend the lines A B and G C in elevation as A P and G S. Take the half diameter L J and place it on either side of *e f* as shown by O P. In a similar manner take the half diameter of the section N as *d i* and place it on either side of *e f* as shown by R S. Then O P S R shows the end elevation. Draw E T intersecting *e f* at T. Now draw the curve O T P, which in this case is struck from the center U, being obtained by bisecting the line O T. It should be understood that the curve O T P, which represents the true section on J E, can be made any desired shape, but when once drawn, represents a fixed line.

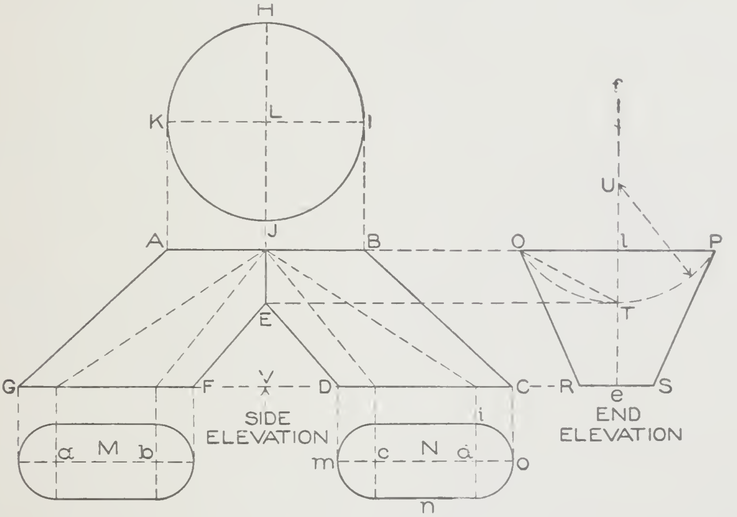


Fig. 102.

The pattern will be developed by triangulation, for which diagrams of sections must be obtained from which to obtain measurements. These sections are obtained as follows: In Fig. 103 1 4 5 12 13 is a reproduction of J B C D E, Fig. 102. Reproduce the quarter profile H L I, the half profile O T, and the half profile *m n o* as shown by 1′ 1 4, 1″ 13 1 and 12 9′ 8′ 5 in Fig. 103. Divide the round ends in *a* each into 3 parts and the profiles *b* and *c* also each into 3 spaces, as shown by the figures. Drop lines from these figures at right angles to the base lines from 1 to 15 as shown and draw solid and dotted lines in the usual manner. While in some of the previous problems only dotted lines were drawn, we have drawn both solid and dotted lines in this case, in order to avoid a confusion of sections. A diagram of sections on solid lines in Fig. 103 is shown in Fig. 104, the figures in both corresponding; while Fig. 105 shows the true sections on dotted lines. The method of obtaining these sections has been described in connection with other problems.

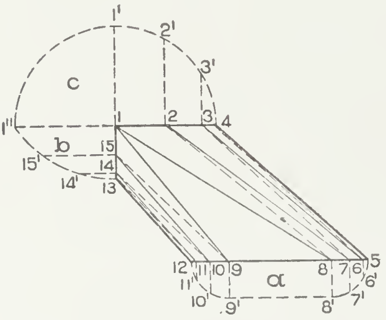


Fig. 103.

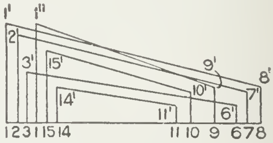


Fig. 104.

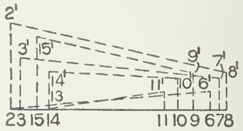


Fig. 105.

For the pattern draw any vertical line as 4 5, Fig. 106, equal to 4 5 in Fig. 103. Then with 5 6′, Fig. 103, as radius and 5 in Fig. 106 as center draw the arc 6, intersecting it by an arc struck from 4 as center and 4 6′, Fig. 105, as radius. Then using 4 3′, Fig. 103, as radius, and 4 in Fig. 106 as center, describe the arc 3, intersecting it by an arc struck from 6 as center and 6′ 3′ in Fig. 104 as radius. Proceed in this manner, using alternately as radii, first the divisions in *a* in Fig. 103, then the slant lines in Fig. 105; the divisions in *c* in Fig. 103, then the slant lines in Fig.

104, until the line 1 8, Fig. 106, is obtained. Now using 8 as center and 8′ 9′, Fig. 103, as radius draw the arc 9 in Fig. 106, intersecting it by an arc struck from 1 as center and 1″ 9′, Fig. 104, as radius. Then starting at 1 in Fig. 106 use alternately as radii, first the divisions in *b* in Fig. 103, then the slant lines in Fig. 105, the divisions in *a* in Fig. 103, then the length of the slant lines in Fig. 104 until the line 12 13 is obtained in Fig. 106, which equals 12 13 in Fig. 103. Trace a line through points thus obtained in Fig. 106, then will 4 1 13 12 9 8 5 be the half pattern. If the pattern is desired in one piece, trace this half opposite the line 4 5 as shown by 1′ 13′ 12′ 9′ 8′, allowing laps for riveting.

In Fig. 107 is shown a perspective view of a tapering flange around a cylinder passing through an inclined roof, the flange being rectangular on the roof line. The problem will be developed by triangulation, a plan and elevation first being required as shown in Fig. 108.

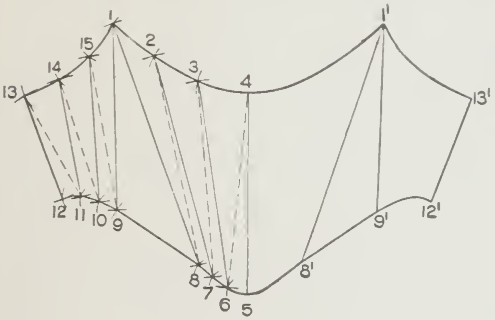


Fig. 106.

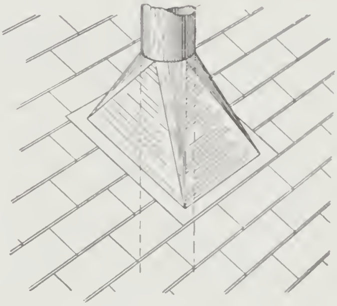


Fig. 107.

First draw the angle of the roof A B at an angle of 45°, through which draw a center line C D. From the roof line A B on the center line set off *a b* equal to 4 inches and through *b* draw the horizontal line E F, making B F and B E each one inch. Through *d* on the center line draw the horizontal line G H, making *d* H and *d* G each two inches. From H and G erect perpendiculars intersecting the roof line at K and L. Then draw lines from E to K and F to L, completing the elevation. Construct the square in plan making the four sides equal to G H. Bisect H I and draw the center line *c e* intersecting the vertical center at *d′*. Then with radius equal to *b* F or *b* E in elevation and *d′* in plan as center.

draw the circle 1 4 7 4′ representing the horizontal section on E F in elevation, while G H I J is the horizontal section on K L in elevation. As the circle in plan is in the center of the square making the two halves symmetrical it is only necessary to divide the semicircle into equal spaces as shown from 1 to 7 and draw lines

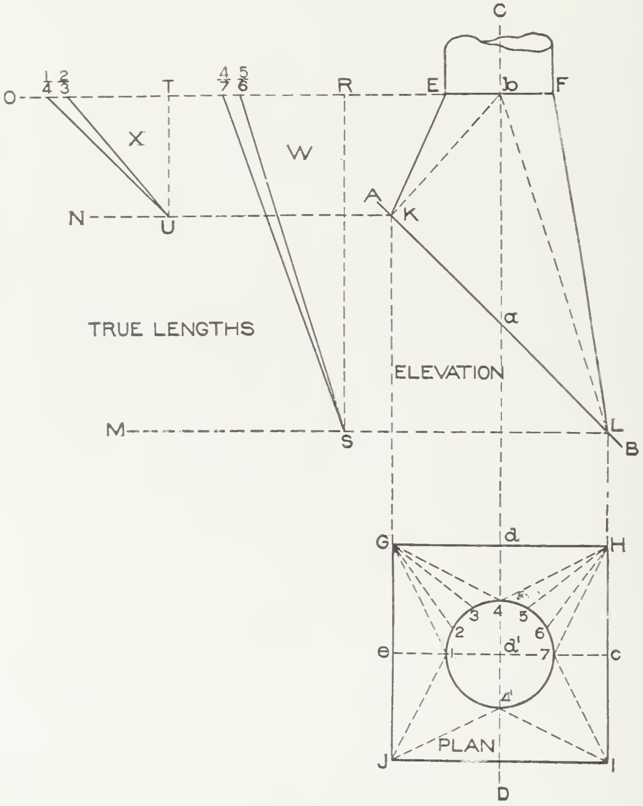


Fig. 108.

from 1, 2, 3 and 4 to G, and 4, 5, 6 and 7 to H. Then will the lines in 1 G.4 and 4 H 7 represent the bases of triangles which will be constructed, whose altitudes are shown respectively by the vertical heights in K E and L F in elevation. Therefore draw horizontal lines through E F, K, and L as shown by F O, K N, and L M. From any point as R and T on F O, draw the perpendiculars R S and T U respectively, meeting the horizontal lines drawn from L and K. Now take the various lengths in plan as G1, G2, G3, and

G4 and place them on the line F O as shown by T1, T2, T3 and T4, from which points draw lines to U which will represent the true lengths on similar lines in plan. In similar manner take the distances in plan from H to 4, to 5, to 6, to 7, and place them on the line F O, from R to 4, to 5, to 6, to 7, from which points draw lines to S which represent the true lengths on similar lines in plan.

For the pattern take the distance F L in elevation and place it on the vertical line 7′ L in Fig. 109. At right angles to 7′ L draw L S equal to *e* H or *e* I in plan. Fig. 108. Draw the dotted line from 7′ to S in Fig. 109, which should be equal to S 7 in W in Fig. 108. Now with radii equal to S $\frac{4}{7}$, and S $\frac{5}{6}$ and S, Fig. 109, as center, draw the arcs indicated by similar numbers. The dividers should equal the spaces in the semicircle in plan in Fig. 108, and starting at 7′ in Fig. 109, step from arc to arc of corresponding numbers as shown by 6′, 5′, 4′. Draw a dotted line from 4′ to S. Then using S as center and L K in elevation, Fig. 108, as radius, describe the arc U in Fig. 109, intersecting it by an arc struck from 4′ as center and U 4, Fig. 108, as radius. Now using U $\frac{1}{4}$, and U $\frac{2}{3}$ in X as radii, and U, Fig. 109, as center, describe arcs having similar numbers. Again set the dividers equal to the spaces in plan in Fig. 108, and starting from 4′ in Fig. 109 step to corresponding numbered arcs as shown by 3′, 2′, 1′.

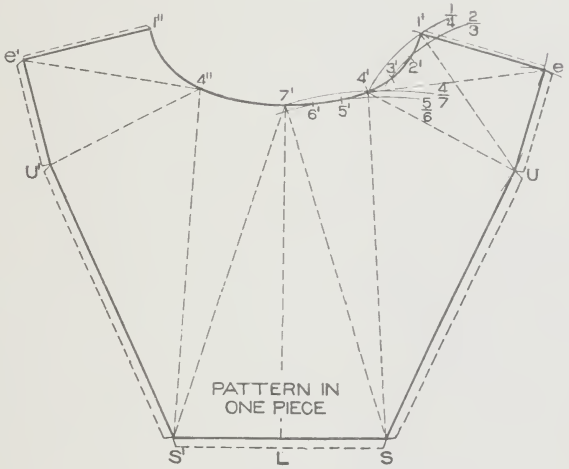


Fig. 109.

Draw a dotted line from 4′ to U to 1′. With K E in elevation, Fig. 108, as radius, and 1′ in Fig. 109 as center, describe the arc *e* intersecting it by an arc struck from U as center and G *e* in plan in Fig. 108 as radius. Draw a line connecting S, U, *e*, and 1′. 7′ 4′ 1′ *e* U S L 7′ shows the half pattern, which can be traced opposite the line 7′ L to complete the full pattern as shown by 7′ 4″ 1″ *e*′ U′ S′ L.

One of the difficult problems often encountered by the sheet metal worker is that of a cylinder joining a cone furnace top at any angle. The following problem shows the principle to be applied, no matter what size the furnace top has, or what size pipe is used, or at what angle the pipe is placed in plan or elevation, the principles being applicable under any conditions.

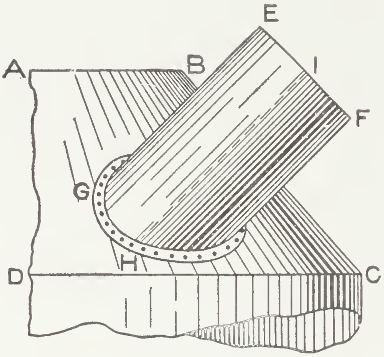


Fig. 110.

Fig. 110 shows a view of a cylinder intersecting a conical furnace top, the pipe being placed to one side of the center of the top. A B C D represents a portion of the conical top, intersected by the cylinder E F G H, the side of the cylinder H I to intersect at a given point on the conical top as at H. This problem presents an interesting study in projections, intersections, and development, to which close attention should be given.

In Fig. 111 first draw the center line A X. Then draw the half elevation A B C D, making A B $1\frac{5}{8}$ inches, C D $3\frac{1}{2}$ inches and the vertical height A D $2\frac{1}{8}$ inches. Draw the line from B to C. Directly below C D draw the one-quarter plan using Z as center, as shown by Z C^1 D^1 and in line with A B of the elevation draw the quarter plan of the top as Z B^1 A^1. Let *a* in the elevation represent the desired distance that the side of the cylinder is to meet the cone above the base line as H in Fig. 110. From *a*, parallel to C D in Fig. 111, draw *a b*. Then from *a* drop a vertical line intersecting the line Z C^1 in plan at *a*′. Then using Z as center and Z *a*′ as radius, describe the quarter circle *a*′ *b*′. Z *a*′ *b*′

in plan represents the true section on the horizontal plane *a b* in elevation. Now locate the point where the side of the cylinder as II in Fig. 110 shall meet the arc *a′ b′* in plan, Fig. 111, as shown

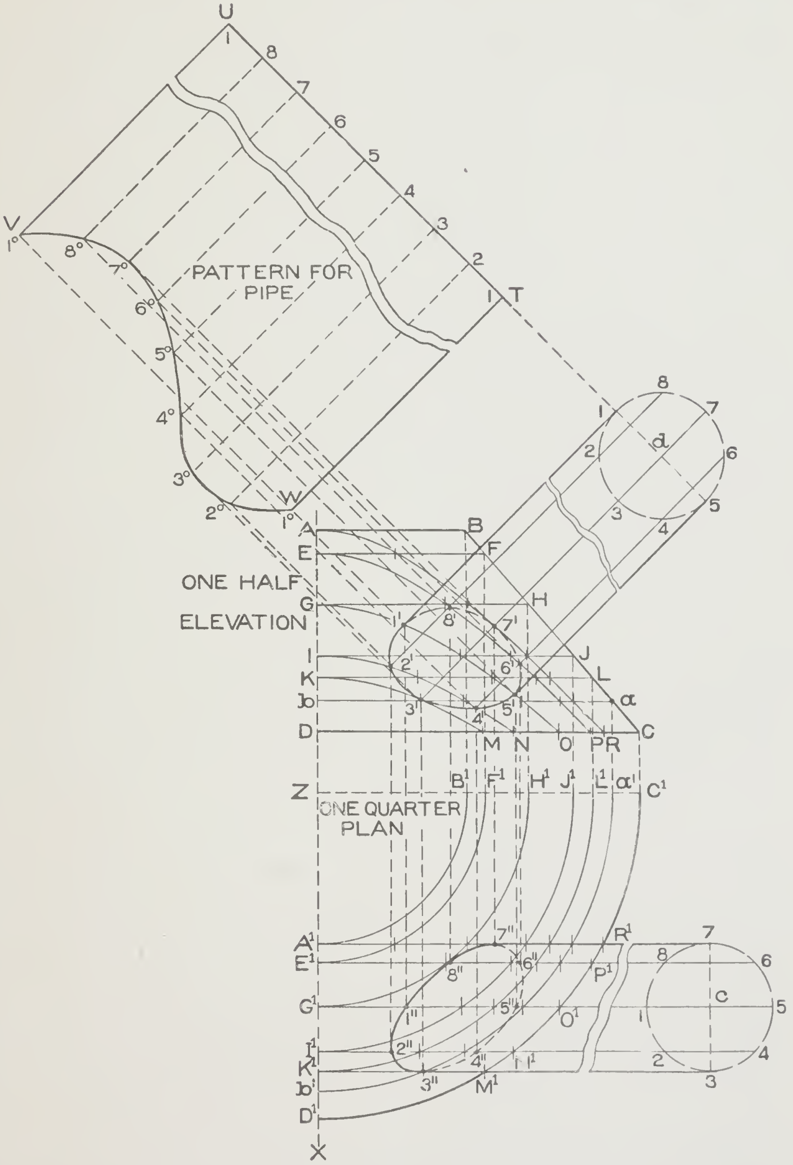


Fig. 111.

at 3″. Through 3″ draw the horizontal line intersecting the center line at K^1, the outer arc at M^1 and extend it indefinitely to 3. From 3 erect the perpendicular equal to the diameter of the cylinder, or $1\frac{5}{8}$ inches, bisect it and obtain the center *c*. Using *c* as center with *c* 7 as radius, describe the profile of the cylinder as shown, and divide it into equal parts from 1 to 8. From these points draw lines parallel to 3 K^1, intersecting the outer arc D^1 C^1 at N^1 O^1 P^1 R^1 and the center line Z X at I^1, G^1, E^1, A^1. With Z as center and the various intersections from K^1 to A^1 as radii, describe the arcs K^1 L^1, I^1 J^1, G^1 H^1, E^1 F^1, and A^1 B^1. From the intersection B^1, F^1, H^1, J^1, L^1 erect vertical lines into the elevation intersecting the side of the cone B C as shown by similar letters B F H J L. From these points draw horizontal lines through the elevation as shown respectively by A B, E F, G H, I J, and K L. These lines represent a series of horizontal planes, shown in plan by similar letters. For example, the arc E^1 F^1 in plan represents the true section on the line E F in elevation, while the arc G^1 H^1 is the true section on the line G H in elevation, etc.

The next step is to construct sections of the cone as it would appear, if cut by the lines shown in plan by K^1 M^1, I^1 N^1, G^1 O^1.E^1 P^1, and A^1 R^1. To obtain the section of the cone in elevation on the line A^1 R^1 in plan, proceed as follows: At right angles to the line A^1 R^1 and from the intersections on the various arcs, draw lines upward (not shown) intersecting similar planes in elevation corresponding to the arcs in plan. A line traced through intersections thus obtained in elevation as shown from A to R, will be the true section on the line A^1 R^1 in plan. For example, the line K^1 M^1 of the cylinder intersects the arcs at K^1 3″ and M^1 respectively. From these intersections, erect vertical lines intersecting K L, *b a*, and D C in elevation at K, 3′, and M respectively. Trace a curve through these points, then will K 3′ M be the section of the cone if cut on the line K^1 M^1 in plan. In similar manner obtain the other sections. Thus the section line E P, G O, and I N in elevation, represent respectively the sections if cut on the lines E^1 P^1, G^1 O^1, and I^1 N^1 in plan. Now from the given point 3″ in plan erect a line which must meet the intersection of the plane *b a* and section K M in elevation at 3′. From 3′ at its desired angle, in this case 45°, draw the line 3′ 7. At any point as *d* at right angles to 3′ 7 draw the line 1 5 through *d*, making *d* 5 and *d* 1 each equal to half the diameter of the cylinder shown in plan. With *d* 5 as radius and *d* as center draw the profile of the cylinder in elevation, and divide it into the same number of parts as shown in C in plan, being careful to allow the circle *d* in elevation to make a quarter turn, bringing the number 1 to the top as shown.

The next operation is to obtain the miter line or line of joint between the cylinder and cone in elevation. By referring to the plan it will be seen that the point 7 in the profile *e* lies in the plane of the section A^1 R^1. Then a line from the point 7 in the profile *d* in elevation, drawn parallel to the lines of the cylinder, must cut the section A R which corresponds to the plane A^1 R^1 in plan as shown by 7′ in elevation. The points 6 and 8 in the profile *e* in plan, are in the plane at the section E^1 P^1, then must the corresponding points 6 and 8 in the profile *d* in elevation, intersect the section E P as shown by 6′ and 8′. As the points 1 5, 2 4, and 3 in the profile *e* in plan, are in the planes of the sections G^1 O^1, I^1 N^1, and K^1 M^1 respectively, the corresponding points 1 5, 2 4, and 3 in the profile *d* in elevation must intersect the sections G O, I N, and K M respectively at points 1′ 5′, 2′ 4′, and 3′ as shown. Trace a line through these points, which will show the line of intersection between the cone and cylinder.

For the pattern for the cylinder, proceed as follows: At right angles to the line of the cylinder in elevation, draw the line T U upon which place the stretchout of the profile *d* as shown by similar figures on T U. In this case the seam of the pipe has been placed at 1 in *d*. Should the seam be desired at 3, 5 or 7, lay off the stretchout on T U starting with any of the given numbers. At right angles to U T from the small figures 1 to 1 draw lines which intersect with lines drawn from similar numbered intersections in the miter line in elevation at right angles to 1′ 1, resulting in the intersections 1° to 5° to 1° in the pattern. Trace a line through points thus obtained, then will U V W T be the development for the cylinder to which laps must be allowed for riveting to the cone as shown in Fig. 110 and seaming the joint T W in pattern in Fig. 111.

While the pattern for the cone is obtained the same as in ordinary flaring ware, the method will be described for obtaining the pattern for the opening to be cut into the cone. Before this can be done a plan view of the intersection between the pipe and cone must first be obtained as follows: From the various intersections 1′ to 8′ in elevation drop vertical lines intersecting lines drawn from similar numbers in the profile *c* in plan, thus obtaining the intersections 1″ to 8″ through which a line is traced which is the desired plan view.

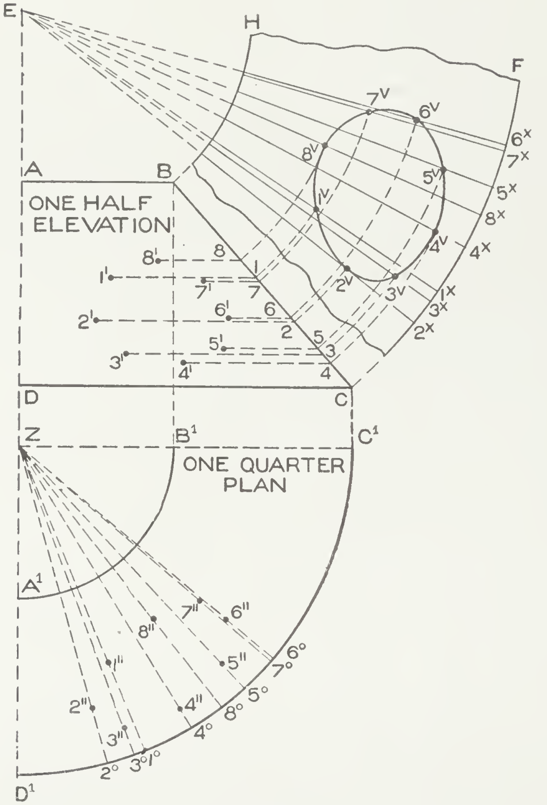


Fig. 112.

For the pattern for the opening in the cone, the outline of the half elevation and one-quarter plan with the various points of intersections both in plan and elevation in Fig. 112 is a reproduction of similar parts in Fig. 111, and has been transferred to avoid a confusion of lines which would otherwise occur in obtaining the pattern. Parallel to D C in Fig. 112 from the various intersections 1′ to 8′ draw lines intersecting the side of the cone B C from 1 to 8. Through the various intersections 1″ to 8″ in plan from the apex Z draw lines intersecting the outer curve from 1° to 8° as shown. Extend the line C B in elevation until it meets the center line D A extended at E. Then using E as center, with E C and E B as radii draw the arcs C F and B H respectively. At any point as 2^X on the arc C F lay off the stretchout of the various points on D^1 C^1 in plan from 2° to 6° as shown by similar figures on C F as shown from 2^X to 6^X. From these points draw radial lines to the apex E, and intersect them by arcs struck from E as center whose radii are equal to the various intersections on B C having similar numbers. Thus arc 4 intersects radial line 4^X at 4^Y; arcs 3, 5, and 2 intersect radial lines 3^X, 5^X, and 2^X at 3^Y, 5^Y, and 2^Y, and so on. Trace a line through points thus obtained as shown from 1^Y to 8^Y which is the desired shape. If a flange is desired to connect with the cylinder, a lap must be allowed along the inside of the pattern.

COPPERSMITH'S PROBLEMS.

In the five problems which will follow, particular attention is given to problems arising in the coppersmith's trade. While all the previous problems given in the course can be used by the coppersmith in the development of the patterns where similar shapes are desired, the copper worker, as a rule, deals mostly with hammered surfaces, for which flaring patterns are required. The principles which will follow, for obtaining the blanks or patterns for the various pieces to be hammered, are applicable to any size or shape of raised work. The copper worker's largest work occurs in the form of brewing kettles, which are made in various shapes, to suit the designs of the different architects who design the work. In hammering large brewing kettles of heavy copper plate, the pieces are developed, hammered, and fitted in the shop, then set together in the building, rope and tackle being used to handle the various sections for hammering, as well as in construction at the building. While much depends upon the skill the workman has with the hammer, still more depends upon the technical knowledge in laying out the patterns.

In all work of this kind the patterns are but approximate, but no matter what size or shape the work has, the principles contained in the following problems are applicable to all conditions.

In Fig. 113 is shown a perspective of a sphere which is to be constructed of horizontal sections as shown in Fig. 114, in which for practice draw the center line A B, on which, using *a* as center, and with radius equal to $2\frac{1}{4}$ inches, describe the elevation of the sphere B C D E. Divide the quarter circle D C into as many spaces as the hemi-sphere is to have sections, as shown by C F G D. From these points draw horizontal lines through the elevation, as shown by C E, F H, and G I. Now through the extreme points as E H, H I, and I D draw lines intersecting the center line B A at J, K, and D respectively. For the pattern for the first section Z, take D I as radius, and using D^1 in Z^1 as center, describe the circle shown. For the pattern for the second section Y, use K I and K H as radii, and with K^1 as center draw the arcs I^1 I^2 and H^2 H^3. From any point as H^3 draw a line to the center K^1. It now becomes necessary to draw a section, from which the true length of the patterns can be obtained. Therefore with *b* F as radius, describe the quarter circle F L, which divide into equal spaces, as shown by the figures 1 to 5. Let the dividers be equal to one of those spaces and starting at H^3 on the *outer arc* in Y^1 step off four times the amount contained in the quarter section F L, as shown from 1

Fig. 113.

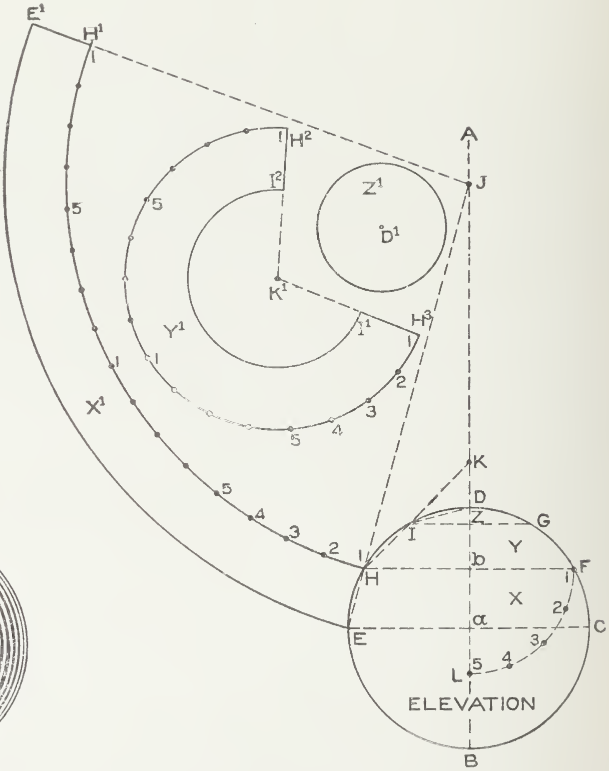


Fig. 114.

to 5 to 1 to 5 to 1 in Y^1. From 1 or H^2 draw a line to K^1. Then will H^2 I^2 I^1 H^3 be the pattern for the section Y in elevation.

For the pattern for the third section, use J as center, and with radii equal to J H and J E draw the arcs H H^1 and E E^1. Now set the dividers equal to one of the equal spaces in F L and starting from H set off four times the amount of L F as shown from 1 to 5 to 1 to 5 to 1 on the *inner curve* H H^1. From the apex J through H^1 draw a line intersecting the outer curve at E^1. E E^1 H^1 H shows the pattern for the center section. It will be noticed in the pattern X^1 we space off on the inner curve, while on the pattern Y^1 we space off on the outer curve. These two curves must contain the same amount of material as they join together when the ball is raised. To all of the patterns laps must be allowed for brazing or soldering. The patterns shown are in one piece; in practice where the sphere is large they are made in a number of sections.

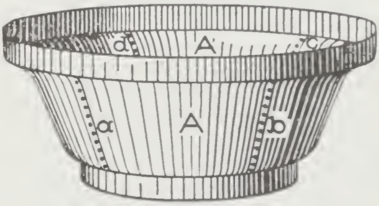


Fig. 115.

In Fig. 115 is shown the perspective view of a circular tank whose outline is in the form of an ogee. The portion for which the patterns will be described is indicated by A A, made in four sections, and riveted as shown by *a b c d*.

Fig. 116 shows how the pattern is developed when the center of the ogee is flaring as shown from 3 to 4 in elevation. First draw the elevation A B C D, making the diameter of A B equal to 7 inches, the diameter of D C 4 inches, and the vertical height of the ogee $1\frac{3}{4}$ inches. Through the center of the elevation draw the center line *f h*, and with any point upon it as *i*, draw the half plan through A B and C D in elevation as shown respectively by E F and H G. Now divide the curved parts of the ogee into equal spaces as shown from 1 to 3 and 4 to 6. Draw a line through the flaring portion until it meets the center line *f h* at *j*. *j* will, therefore, be the center with which to strike the pattern. Take the stretchout of the curve from 3 to 1 and 4 to 6 and place it on the flaring line from 3 to 1′ and 4 to 6′ as shown by the figures. Then will 1′ 6′ be the stretchout for the ogee. It should be understood that no hammering is done to that part shown from 3 to 4. The portion shown from 3 to 1′ is stretched to meet the required profile 3 2 1, while the lower part 4 to 6′ is raised to conform with the lower curve 4 5 6. Therefore, knowing that the points 3 and 4 are fixed points, then from either of these, in this case point 4,

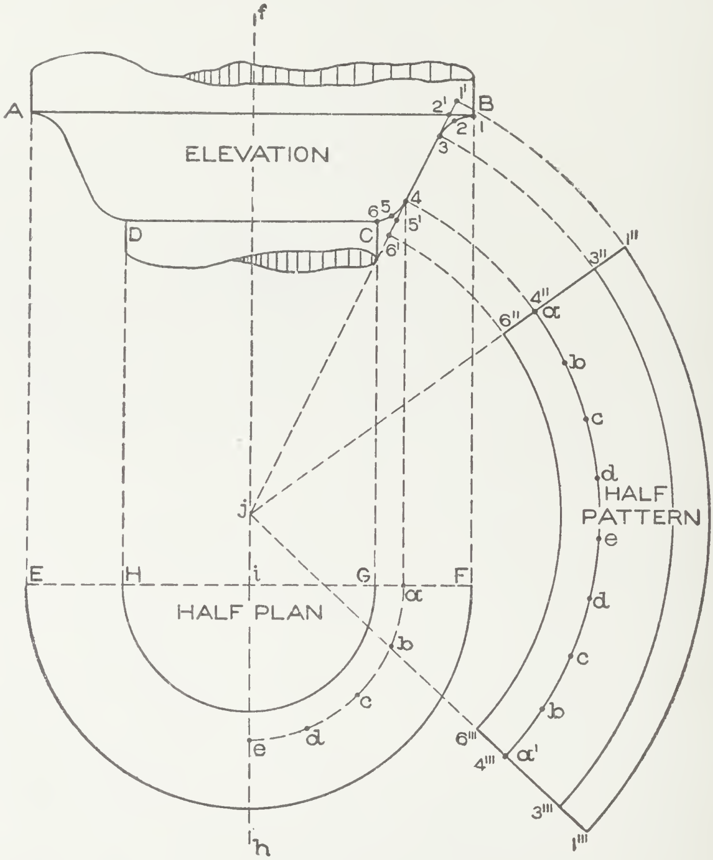


Fig. 116.

drop a vertical line intersecting the center line E F in plan at *a*. Then with *i* as center and *i a* as radius, describe the quarter circle *a e*, and space it into equal parts as shown by *a*, *b*, *c*, *d*, *e*, which represent the measuring line in plan on the point 4 in elevation. Using *j* as center, and *j* 6′, *j* 4, *j* 3 and *j* 1′ as radii, draw the arcs 1″-1‴, 3″-3‴, 4″-4‴ and 6″-6‴ as shown. From 1″ draw a radial line to *j* intersecting all the arcs as shown. Now starting at 4″ step off on the arc 4″-4‴ twice the stretchout of the quarter circle *a e* as shown by similar letters *a* to *e* to *a′* in pattern. From *j* draw a line through *a′* intersecting all of the arcs as shown. 1″-1‴-6‴-6″ shows the half pattern for the ogee.

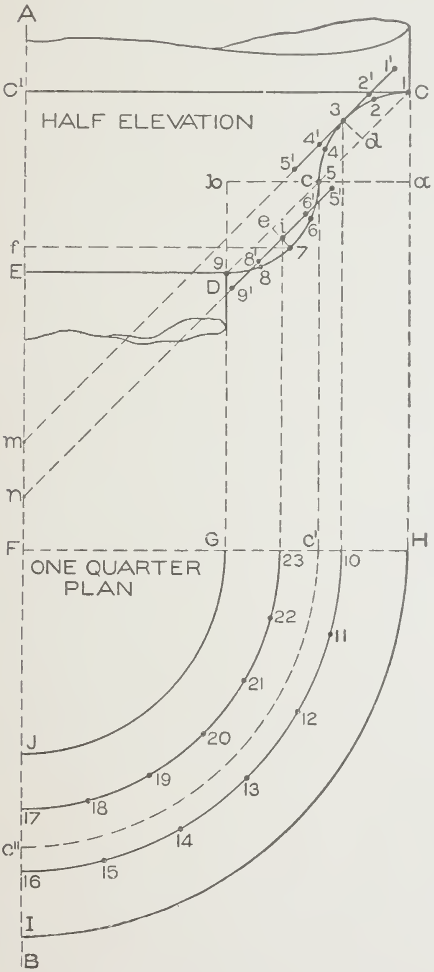


Fig. 117.

While in the previous problem the greater part of the ogee was flared, occasion may arise where the ogee is composed of two quarter circles struck from centers as shown in Fig. 117. First draw the center line A B, then draw the half diameter of the top C^1 C equal to $3\frac{1}{4}$ inches and the half diameter E D $1\frac{3}{4}$ inches. Make the vertical height of the ogee $1\frac{1}{2}$ inches, through the center of which draw the horizontal line *a b*. From C and D draw vertical lines intersecting the horizontal line *a b*, at *a* and *b* respectively. Then using *a* and *b* as centers with radii equal respectively to *a* C and *b* D draw the quarter circles shown completing the ogee. In the quarter plan below which is struck from the center F, G J and H I are sections respectively on D E and C C^1 in elevation. The methods of obtaining the patterns in this case are slightly different than those employed in the previous problems. The upper curve shown from C to *c* will have to be stretched, while the lower curve shown from *c* to D will have to be raised. Therefore in the stretchout of the pattern of the upper part from 1′ to 3 and 3 to 5′ the edges must be stretched so as to obtain more material to allow the metal to increase in diameter and conform to the desired shape shown from 1 to 3 and 3 to 5. In the lower curve the opposite method must be employed. While in the upper curve the edges had to be stretched to increase the diameters, in the lower curve the edges must be drawn in by means of raising, to decrease the diameter, because the diameters to the points 5″ and 9′ are greater than to points *c* and *d*.

To obtain the pattern for the upper curve C *c* which must be stretched, draw a line from C to *c*; bisect it and obtain *d*, from which erect the perpendicular *d* 3 intersecting the curve at 3. Through 3 draw a line parallel to C *c* intersecting the center line A B at *m*. Now divide the curve C *c* into equal spaces as shown from 1 to 5 and starting from the point 3 set off on the line just drawn on either side of 3 the stretchout shown from 3 to 1′ and 3 to 5′. 1′ 5′ shows the amount of material required to form the curve C *c*. In this case 3 represents the stationary point of the blank on which the pattern will be measured. Therefore from 3 drop a vertical line intersecting the line F H at 10. Then using F as center and F 10 as radius, describe the arc 10 16, and divide it into equal spaces as shown from 10 to 16. Now with radii equal to *m* 5′, *m* 3 and *m* 1′, Fig. 117, and with *m* in Fig. 118 as center, describe the arcs 5 5′, 3 3′ and 1 1′. Draw the radial line *m* 1 intersecting the two inner arcs at 3 and 5. As the arc 3 3′ represents the stationary point 3 in elevation in Fig. 117, then set the dividers equal to the spaces 10 16 in plan and step off similar spaces in Fig. 118 on the arc 3 3′, starting at 3 as shown by similar numbers 16 to 10. Through 10 draw a line to the apex *m*, intersecting the inner curve at 5′ and the outer curve at 1′. 1 1′ 5′ 5 is the quarter pattern for the upper curve or half of the ogee, to which laps must be allowed for riveting and brazing.

For the pattern for the lower curve in elevation in Fig. 117 draw a line from *c* to D; bisect it at *e* and from *e* erect a perpendicular intersecting the curve at 7. From 7 draw a horizontal line intersecting the center line at *f*. Now the rule to be followed in "raising" is as follows: Divide the distance from *e* to 7 into as many parts, as the half diameter F 7 is equal to inches. In this case 7 *f* equals $2\frac{1}{4}$ inches; (any fraction up to the $\frac{1}{2}$ inch is not taken into consideration, but over ½ inch one is added). Therefore for 2¼ inches use 2. Then divide the distance from *e* to 7 into two parts as shown at *i* and through *i* parallel to *e* D draw a line as shown intersecting the center line at N. Now divide the curve *e* to D into equal spaces as shown by the figures 5 to 9. Let off on either side of *i* the stretchout from 5 to 9 as shown from 5″ to

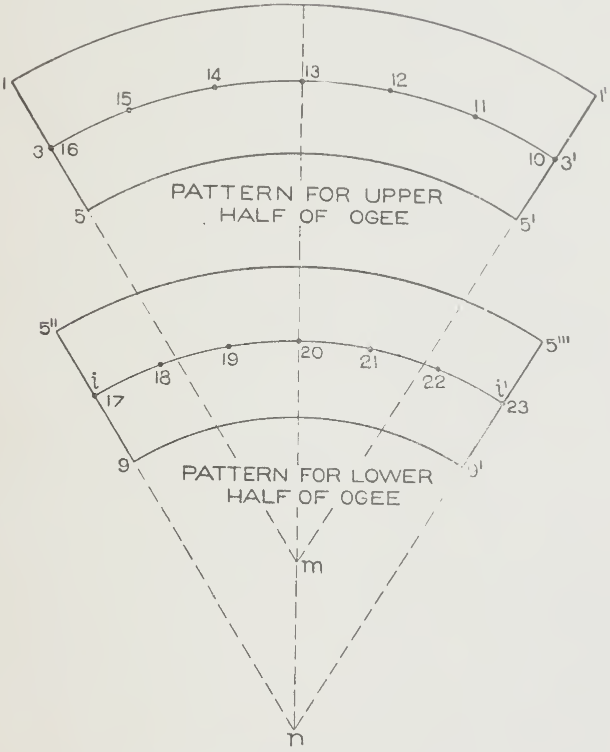

1
15
14
13
12
11
1'
3
16
10
3'
PATTERN FOR UPPER HALF OF OGEE
5
5'
5''
5'''
i
18
19
20
21
22
i'
17
23
9
9'
PATTERN FOR LOWER HALF OF OGEE
m
n

Fig. 118.

9′. From *i* drop a vertical line intersecting F H in plan at 23. Then using F as center draw the arc 23 17 as shown, which represents the measuring line in plan on *i* in the stretchout.

The student may naturally ask, why is *i* taken as the measuring line in plan, when it is not a stationary point, for when "raising" *i* will be bulged outward with the raising hammer until it meets the point 7. In bulging the metal outward, the surface at *i* stretches as much as the difference between the diameter at *i* and

7. In other words, if the measuring point were taken on 7 it would be found that after the mould was "raised" the diameter would be too great. But by using the rule of dividing e 7 into as many parts as there are inches in f 7 the diameter will be accurate while this rule is but approximate. In this case e 7 has only been divided into two equal parts, leaving but one point in which a line would be drawn through parallel to c D. Let us suppose that the semi-diameter 7 f is equal to eleven inches. Then the space from e to 7 would be divided into just so many parts, *and through the first part nearest the cove* the line would be drawn parallel to c D and used as we have used i. Now with radii equal to n 9′, n i, and n 5″ and n in Fig. 118 as center, describe the arcs 5″ 5‴ i i' and 9 9′. From any point as 5″ draw a line to n intersecting all the arcs shown. Now take the stretchout from 17 to 23 in plan, Fig. 117, and starting from 17 in Fig. 118 mark off equivalent distances on the arc i i' as shown. Draw a line through 23 to the apex n, intersecting the inner and outer arcs at 9′ and 5‴. Then will 9 5″ 5‴ 9′ be the greater pattern for the lower part of the ogee.

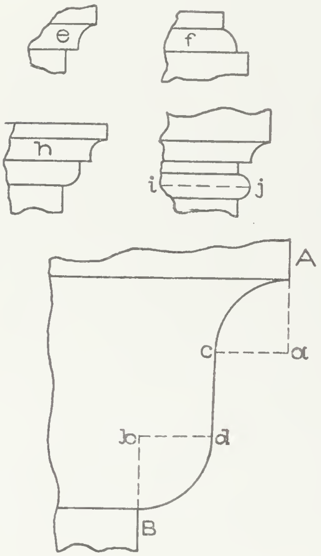


Fig. 119.

Another case may arise where the center of the ogee is vertical as shown from c to d in Fig. 119 in A B. In this case the same principles are applied as in Fig. 117; the pattern for c d in Fig. 119 being a straight strip as high as c d and in length equal to the quarter circumference c' c'' in plan in Fig. 117 which is the section on c in elevation. These rules are applicable to any form of mould as shown in Fig. 119, by e, f, h, and j. The bead i in j would be made in two pieces with a seam at i as shown by the dotted line, using the same method as explained in connection with c D in elevation in Fig. 117.

The coppersmith has often occasion to lay out the patterns for curved elbows. While the sheet metal worker lays them out in pieces, the coppersmith's work must form a curve as shown in Fig. 120 which represents a curved elbow of 45°.

In Fig. 121 is shown how an elbow is laid out having 90°, similar principles being required for any degree of elbow. First draw the side elevation of the elbow as shown by A B C D, making the radius E B equal to 4½ inches and the diameter B C 2 inches. Bisect C B at K. Then with E as center and E K as radius draw the arc K J representing the seam at the sides. Draw the front view in its proper position as F G H, through which draw the center line F I representing the seam at back and front, thus making the elbow in four pieces. Directly below C B draw

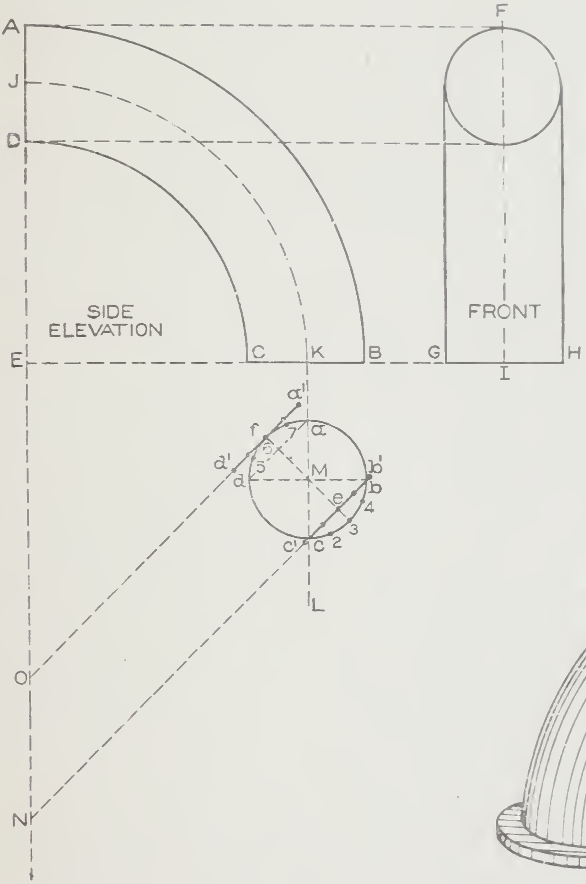


Fig. 121.

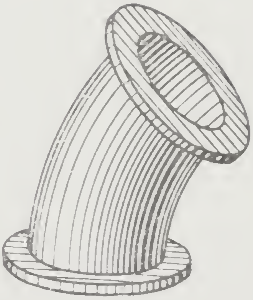


Fig. 120.

the section of the elbow as shown by *a b c d* struck from M as center. Through M draw the diameters *b d* and *a c*. The inner curve of the elbow *a d c* in plan will be stretched, while the outer curve *a b c* in plan will be raised. Through M draw the diagonal 3 6 intersecting the circle at 3 and *f* respectively. Now draw *a d;* through *f* parallel to *a d* draw a line intersecting the center

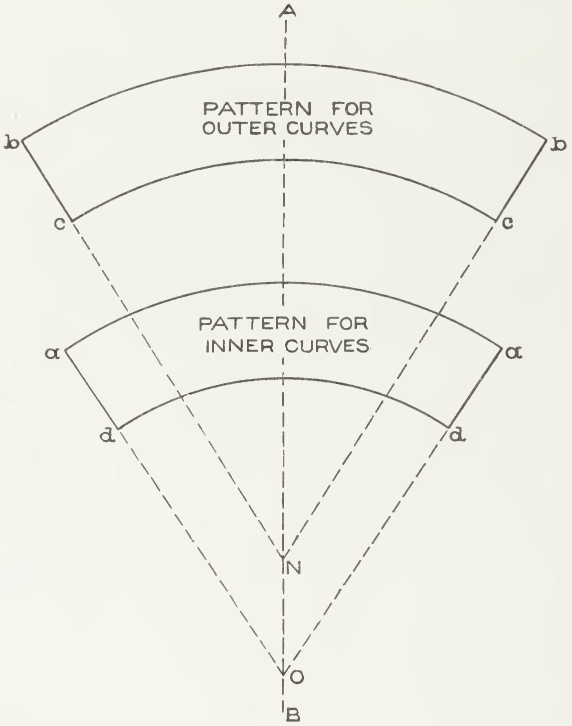


Fig. 122.

line A E extended at O. On either side of *f* place the stretchout of 6 *a* and 6 *d* as shown by *f a'* and *f d'*. Then with radii equal to O *d'* and O *a'* and with O on the line A B, Fig. 122, as center describe the arcs *d d* and *a a*. Make the length of *d d* equal to the inner curve D C in Fig. 121. From *a* and *d* in Fig. 122 draw lines to the apex O extending them to meet the outer curve at *a* and *a*. Then will *a d d a* be the half pattern for the inner portion of the elbow for two sides. The radius for the pattern for the outer curve is shown in Fig. 121 by N *c'*, N *b'*, placing the stretchout of the curve on either side of the point *e*. *b b e e* in Fig. 122 shows the pattern for the outer curve, the length *b b* being obtained from A B in elevation in Fig. 121.

In work of this kind the patterns are made a little longer, to allow for trimming after the elbow is brazed together. Laps must be allowed on all patterns for brazing.

Fig. 123 shows a perspective view of a brewing kettle, made in horizontal sections and riveted. The same principles which were employed for obtaining the patterns for a sphere in Fig. 114 are applicable to this problem. Thus in Fig. 124, let A B C represent a full section of a brewing kettle as required according to architect's design. Through the middle of the section draw the center line D E. Now divide the half section B to C into as many parts as the kettle is to have pieces as shown by *c*, *d*, *e*, *f*. From these small letters draw horizontal lines through the section, as shown by *c* A, *d d′*, *e e′*, and *f f′* and in its proper position below the section. draw the plan views on each of these horizontal lines in elevation, excepting *d′ d*, as shown respectively by I F G H, *e″ e‴* and *f‴ f⁗*, all struck from the center *a*. Now through the points *c d* draw a line which if extended would meet the center line. Then this intersection would be the center with which to draw the arcs *c c′* and *d d″*; the flange *c b* would be added to the pattern as shown by *b′*. The stretchout for this pattern 1^1 would be obtained from the curved line F G H I in plan and stepped off on the outer arc *c c′*. In similar manner through *d e*, *e f*, and *f* C draw the lines intersecting the center line D E at K, L, and C. Then using the points as center, describe the patterns 2^1, and 3^1, and the full circle 4^1.

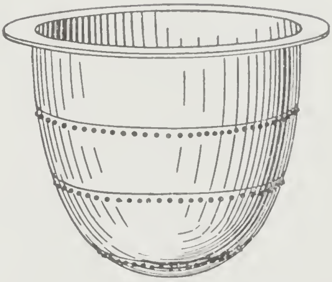


Fig. 123.

The stretchout for the patterns 2^1 and 3^1 is obtained from the circle *e″ e‴* in plan and placed on the inner curve of the pattern 2^1, and on the outer curve of the pattern 3^1. If desired the stretchout could be taken from *f″ f‴* in plan, and placed on the inner curve of 3^1 which would make the pattern similar as before.

In large kettles of this kind, the length of the pattern is guided by the size of the sheets in stock, and if it was desired that each ring was to be made in 8 parts then the respective circle in plan from which the stretchout is taken would be divided into 8 parts, and one of these parts transferred to the patterns, to which laps must be allowed for seaming and riveting.

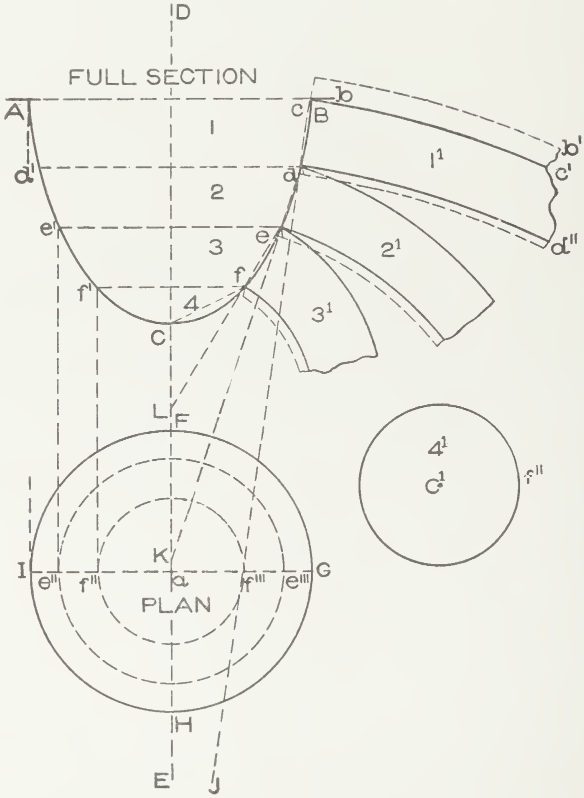


Fig. 124.

PROBLEMS FOR WORKERS IN HEAVY METAL.

While all of the problems given in this course are applicable to developments in heavy metal as well as in that of lighter gauge, the following problems relate to those forms made from boiler plate.

When using metal of heavier gauge than number 20, for pipes, elbows, or any other work, it is necessary to have the exact inside diameter. It is customary in all shops working the heavier metal, to add a certain amount to the stretchout to make up for the loss incurred in bending, in order that the inside diameter of the article (pipe, stack, or boiler shell) may be kept to a uniform and desired size. This amount varies according to different practice of workmen, some of whom allow 7 times the thickness of the metal used, while others add but 3 times the thickness. Theoretically the amount is 3.1416 times the thickness of the metal.

For example, suppose a boiler shell or stack is to be made 48 inches in diameter out of $\frac{1}{2}$-inch thick metal. If this shell is to measure 48 inches on the inside, add the thickness of the metal, which is $\frac{1}{2}$ inch, making $48\frac{1}{2}$ inches. Multiply this by 3.1416 and the result will be the width of the sheet. If, on the other hand, the outside diameter is to measure 48 inches, subtract the thickness of the metal, which would give $47\frac{1}{2}$ inches and multiply that by 3.1416 which would give the proper width of the sheet. It is well to remember that no matter what the thickness of the plate may be, if it is not added, the diameter of the finished article will not be large enough; for where no account is taken of the thickness of the metal, the diameter will measure from the center of the thickness of the sheet. While this rule is theoretically correct there is always a certain amount of material lost during the forming operations. It is, therefore, considered the best practice to use seven times the thickness of the metal in question. The circumference for a stack 48 inches in diameter inside using $\frac{1}{2}$ inch metal would be, on this principle, $3.1416 \times 48 + (7 \times \frac{1}{2})$ to which laps would have to be allowed for riveting. Where the stack has both diameters equal a butt joint is usually employed with a collar as shown at either *a* or *b* in Fig. 125, but where one end of the stack is to fit into the other, a tapering pattern must be obtained which will be described as we proceed.

In putting up large boiler stacks it is usual to finish at the top with a moulded cap, and while the method of obtaining the patterns is similar to parallel line developments, the method of developing such a pattern will be given showing how the holes are punched for a butt joint.

In Fig. 126 a view of the moulded cap on a stack is shown. On a large size stack the cap is often divided into as many as 32 pieces. If the stack is to be made in horizontal sections the rules given in the problems on coppersmithing apply. While in obtaining the patterns for a cap in vertical sections, the plan is usually divided into 16 to 32 sides, according to the size of the stack; we have shown in Fig. 127 a quarter plan so spaced as to give 8 sides to the full circle. This has been done to make each step distinct, the same principles being applied no matter how many sides the plan has.

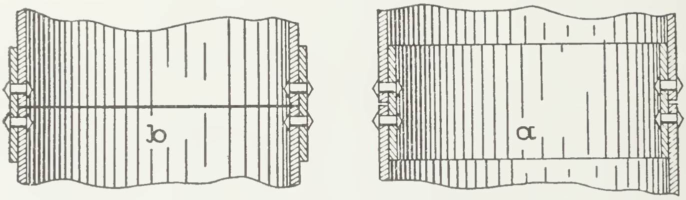


Fig. 125.

First draw the center line A B and with any point as C with radius equal to $4\frac{1}{2}$ inches draw the quadrant D E. Now tangent to D and E, draw the line D F and E G, and at an angle of 45°, tangent to the curve at Y, draw G F intersecting the previous lines drawn at G and F. C D F G E shows the plan view of the extreme outline of the cap. Directly above the plan draw a half section of the cap, the curve 5 8 being struck from *b* as center and with a radius equal to *b* 8 or $1\frac{3}{4}$ inches. Then using the same radius with *a* as center describe the quarter circle 5 2. Make 2 1 equal to $\frac{5}{8}$ inch, and 8 9 one inch. From the corners F and G in plan draw the miter lines F C, C G. Divide the profile of the cap into equal spaces as shown by the figures 1 to 9, from which drop vertical lines, intersecting the miter line F C as shown. On C D extended as C H place the stretchout of the profile of the cap as shown by similar numbers. At right angles to D H draw lines as shown, and intersect them by lines drawn parallel to D H from the intersections on C F. Trace a line through points thus obtained as shown by J I and trace this outline on the opposite side of the

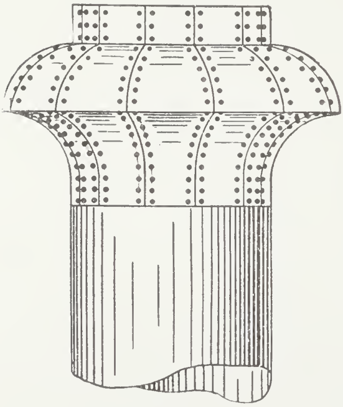


Fig. 126.

line D H as shown by J^1 I^1. Then will J I I^1 J^1 be the complete pattern for one side.

When riveting these pieces together an angle is usually placed on the inside and the miters butt sharp, filing the corners to make a neat fit. This being the case the holes are punched in the pattern before bending as shown by X X X etc. Assuming that the stack on which the cap is to fit is 48 inches in diameter, obtain the circumference as previously explained and divide by 8 (because the plan is composed of 8 pieces) placing one-half of the distance on either side of the center line D H in pattern. Assuming that $\frac{1}{16}$ of the circumference is equal to 9 e, trace from e the entire miter cut, as partly shown by e i to the line I^1 I. If the $\frac{1}{16}$ circumference were equal to 9 d, the cut would then be traced as shown in part by d h until it met the line I I^1. This, of course,

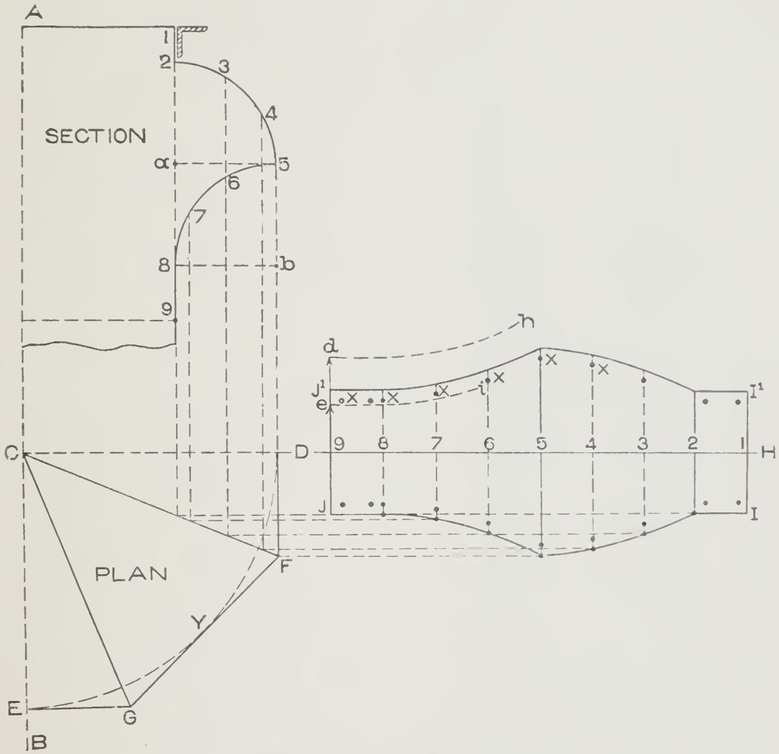


Fig. 127.

would be done on the half pattern 9 J I I before tracing it opposite the center line D H. Should the plan be divided into 32 parts, divide the circumference of the stack by 32 and place $\frac{1}{64}$ of the circumference on 9 J in pattern, measuring from the center line D H, and after obtaining the proper cut, trace opposite the line D H.

In constructing a stack where each joint tapers and fits inside of the other, as shown in Fig. 128, a short rule is employed for obtaining the taper joints without having recourse to the center. In the illustration *a b* represents the first joint, the second C slipping over it with a lap equal to *f*, the joint being riveted together at *e* and *d*. When drawing the first taper joint *a b*, care must be taken to have the diameter at *f* on the outside, equal to the inside diameter at the bottom at *h*. This allows the second joint to slip over a certain distance so that when the holes are punched in the sheets before rolling, the holes will fit over one another after the pipe is rolled.

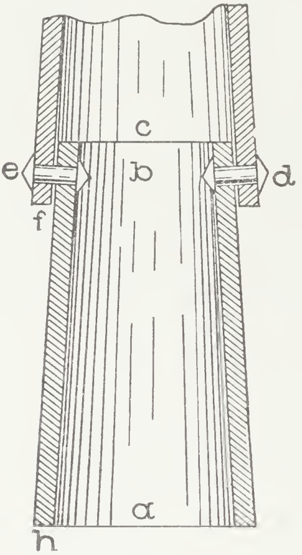


Fig. 128.

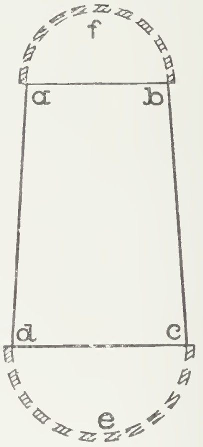


Fig. 129.

In Fig 129 *a b c d* is a taper joint drawn on the line of its inside diameter, as explained in Fig. 128 *f*, and *e* in Fig 129 represents respectively the half sections on *a b* and *d c*. By the short rule the radial lines of the cone are produced without having recourse to the apex, which, if obtained in the full-size drawings, would be so far away as to render its use impracticable. A method similar to the following is used for obtaining the arcs for the pattern in all cases where the taper is so slight as to render the use of a common apex impracticable.

Let *a b c d*, Fig. 130, be a reproduction of *a b c d* in Fig. 129. On either side of *a d* and *b c*, in Fig. 130, place duplicates of *a b c d* as shown by *b′ c′* and *a′ d′*. This can be done most accurately by using the diagonals *d b* and *c a* as radii, and with *d* and *c* as centers describe the arcs *b b′* and *a a′* respectively, and intersect

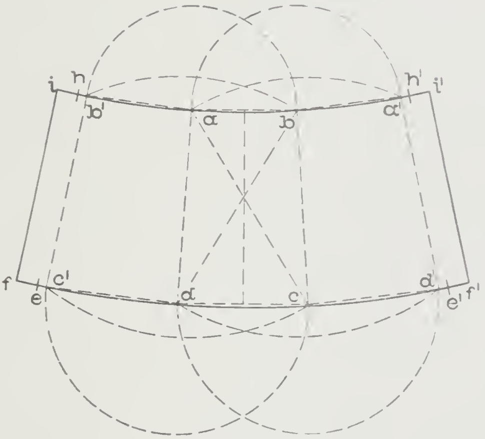


Fig. 130.

them by arcs struck from *a* and *b* as centers, with radii equal respectively to *a b* and *b a* as shown. In precisely the same manner obtain the intersection *c′* and *d′* at the bottom. Now through the intersections *b′ a b a′* and *d′ c d c′* draw the curve as shown by bending the straight-edge or any straight strip of wood placed on edge and brought against the various intersections, extending the curves at the ends and top and bottom indefinitely. Since the circumference of the circle is more than three times the diameter, and as we only have three times the diameter as shown from *c′* to *d′* and *b′* to *a′*, then multiply .1416 times the bottom and top diameter *d c* and *a b* respectively, and place one-half of the amount on either side of the bottom and top curves as shown by *e*, *e′*, and *h*, *h′*. Now take one-half of seven times the thickness of the metal in use and place it on either side on the bottom and top curves as shown by f, f' and i, i', and draw a line from i to f and i' to f'. To this lap must be allowed for riveting. The desired pattern is shown by $i\ i'\ f'\ f$.

Fig. 131 shows a three-pieced elbow made from heavy metal, the two end pieces fitting into the center pieces, to which laps are allowed for riveting. The principles which shall be explained to cut these patterns and make the necessary allowance for any thickness of metal is applicable to any elbow.

In Fig. 132 draw as previously described the elbow A B C, below G H draw the section of the inside diameter as D which is struck from a, and divide into equal spaces as shown by the figures 1 to 5 on both sides. Through these figures draw vertical lines intersecting the miter line $b\ c$, and from these intersections parallel to $c\ d$ draw lines intersecting the line $d\ e$ as shown.

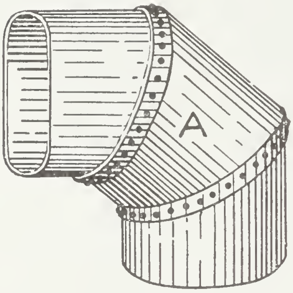


Fig. 131.

Before obtaining the stretchout for these elbows, a preliminary drawing must be constructed, in which an allowance is made for the thickness of the material that is to be used. This drawing makes practical use of a principle well known to draughtsmen from its application to the proportional division of lines and is clearly shown at (R). In allowing for the thickness of the metal in use, it is evident that we cannot allow it at one end, but must distribute it uniformly throughout the pattern. In (R) draw any horizontal line as E F, upon which place the stretchout of the inside diameter of the pipe D, as shown by similar figures on E F. From 1° on E F lay off the distance $1^{\circ}\ m$ equal to 7 times the thickness of the metal in use as before explained. Then using E as center and E m as radius, draw the arc $m\ 1'$ intersecting the vertical line drawn from 1°, and from the various intersections from 1 to 1° on E F erect perpendiculars intersecting the slant line 1 1′ at 2′ 3′ 4′, etc., as shown. The slant line 1 1 with the various intersections is now the correct stretchout for the elbow made of such heavy material called for by the specifications. On G H extended, as H I, place the stretchout of the slant line 1 1′ as shown from 1 to 1′ on H I. At right angles to H I and from the various intersections, erect lines, which are intersected by lines drawn parallel to H I from similar numbered intersections on the miter line *b c*. Trace the curve L M. L M I H shows the pattern for the two end pieces of the elbow.

As the middle section A in Fig. 131 is to overlap the two end pieces, it is unnecessary to allow for any additional thickness on account of this lap when suitable flanging machines are available; but since it is desirable, in some instances, to make an allowance in the pattern for riveting, the method of allowing for this lap will be explained.

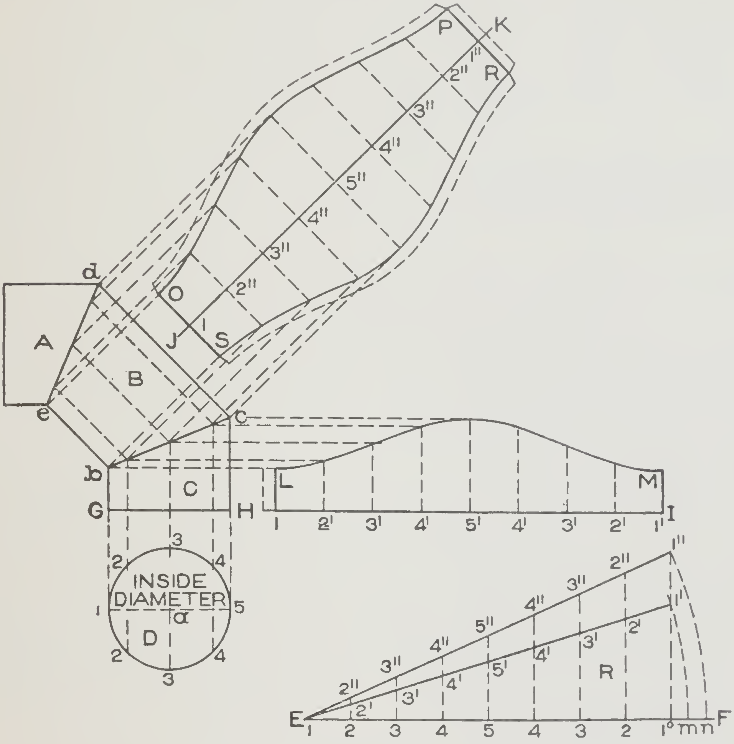


Fig. 132.

In (R), Fig. 132, lay off on the line E F the distance *m n* equal to 7 times the thickness of the metal in use, and with radius equal to E *n* draw an arc intersecting the line 1° 1′ extended at 1″. Draw the slant line from 1″ to 1 and extend all the vertical lines to intersect 1 1″ at 2″ 3″ 4″, etc. The slant line 1 1″ is the correct stretchout for the middle section B. At right angles to *d c* draw J K equal to 1 5″ 1″ in (R), as shown by similar figures in J K, through which draw lines at right angles to J K, and intersect them by lines drawn at right angles to *d c* as shown. Trace the curved lines to produce O P R S, which is the pattern for the middle section, to which flanges are allowed as shown by dotted lines.

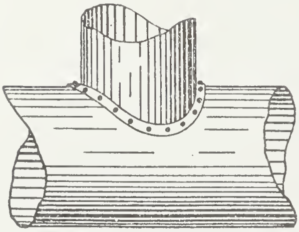


Fig. 133.

The perspective of an intersection between pipes having different diameters in boiler work is shown in Fig. 133. While the method of obtaining the patterns is similar in principle to parallel line developments, a slight change is required in obtaining the allowance in the stretchout for the thickness of the metal in use.

Let A B, Fig. 134, represent the part section of a boiler struck with a radius equal to $3\frac{3}{8}''$ and let 1 7 7° 1° be the elevation of the intersecting pipe, whose inside diameter is $4\frac{7}{8}''$, as shown by 1 7.

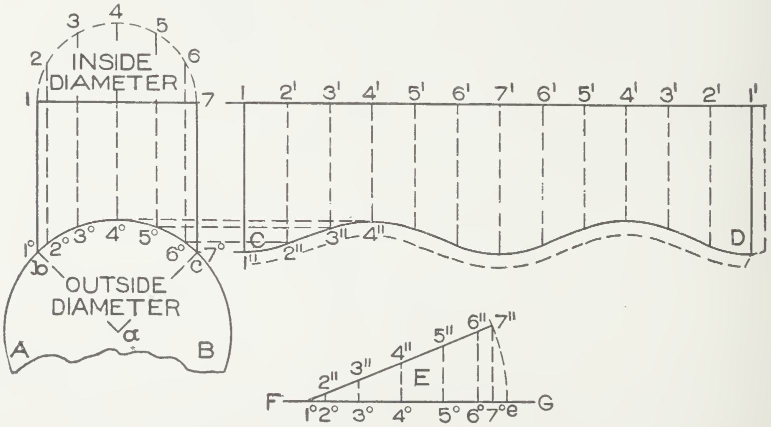


Fig. 134.

Divide the half section 1 4 7 into an equal number of spaces, as numbered, from which drop vertical lines intersecting the outside line of the boiler at 1° to 7° as shown. A true stretchout must now be obtained in which allowance has been made for the thickness of the metal in use. Therefore, in Fig. 135, on the horizontal line A B lay off the stretchout of twice the inside section of the pipe in Fig. 134, as shown by similar figures on A B in Fig. 135, adding $1^x a$, equal to 7 times the thickness of the metal in use. For example, supposing $\frac{1}{4}$-inch steel was used; the distance $1^x a$ would then be equal to 7 × $\frac{1}{4}$, or $1\frac{3}{4}$ inches. Now draw the arc a 1′, using 1 as center, which is intersected by the vertical line drawn from 1^x. From 1′ draw a line to 1, and from the various points on A B erect perpendiculars intersecting 1 1′ at 2′ 3′ 4′, etc. 1 1′ shows the true stretchout to be be laid off on the line 1 7 extended in Fig. 134 as 1 1′, and from the various intersections on 1 1′ drop vertical lines and intersect them by lines drawn parallel to 1 1′ from similar intersections on the curve 1° 7° as shown. Trace a curved line as shown from C to D. 1 C D 1′ shows the pattern for the vertical pipe to which a flange must be allowed for riveting as shown by the dotted line.

It is now necessary to obtain the pattern for the shape to be cut out of the boiler sheet, to admit the mitering of the vertical pipe. In some shops the pattern is not developed, only the vertical pipe is flanged, as shown in Fig. 133, then set in its proper position on the boiler and line marked along the inside diameter of the pipe, the pipe is then removed and the opening cut into the boiler with a chisel. We give, however, the geometrical rule for obtaining the pattern, and either method can be used.

As A B in Fig. 134 represents the outside diameter of the boiler, to which 7 times the thickness of the metal used must be added to the circumference in laying out the sheet, and as the vertical pipe intersects one-quarter of the section as shown by a b c, take the stretchout from 1° to 7° and place it from 1° to 7° on F G in (E), to which add 7° e, equal to $\frac{1}{4}$ of 7 times the thickness of the plate used. Draw the arc e 7″, using 1° as center, intersecting it by the vertical line drawn from 7°. Erect the usual vertical lines and draw 7″ 1°, which is the desired stretchout. Now place this stretchout on the line A B in Fig. 136, erecting vertical lines as shown. Measuring in each and every instance from the line 1 7 in Fig. 134, take the various distances to points 2, 3, 4, 5, and 6 and place them in Fig. 136 on lines having similar numbers, measuring in each instance from A B on either side, thus obtaining the points 2, 3, 4, 5, and 6. Trace the curve 1° 4 7″ 4, which is the desired shape.

Fig. 137 shows a perspective of a gusset sheet A on a locomotive, the method of obtaining this pattern in heavy metal is shown in Fig. 138. First draw the end view A B C, the semicircle 4 1 4 being struck from *a* as center with a radius equal to 2 inches. Make the distance 4 to C and 4 to B both $3\frac{3}{4}$ inches and draw C B. Draw the center line A F, on which line measure up $2\frac{1}{4}$ inches and obtain *b*, which use as center with radius equal to *a* 4, draw the section of the boiler D E F G. In its proper position draw the side view H I J K L M N. H I L M N H shows the side view of the gusset sheet shown in end view by G A E D G.

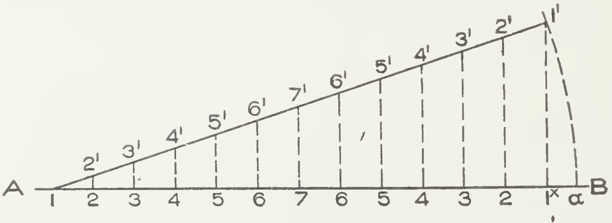


Fig. 135.

Divide the semicircle 4 1 4 in end view into equal spaces as shown, from which draw horizontal lines intersecting H N in side view from 1′ to 4′. From these intersections parallel to H I, draw lines indefinitely intersecting I L from 1″ to 4″. At right angles to N L produced draw the line at *c d*, on which a true section must be obtained at right angles to the line of the gusset sheet. Measuring from the line A D in end view, take the various distances to points 2, 3, and 4 and place them on corresponding lines measuring from the line *c d* on either side, thus obtaining

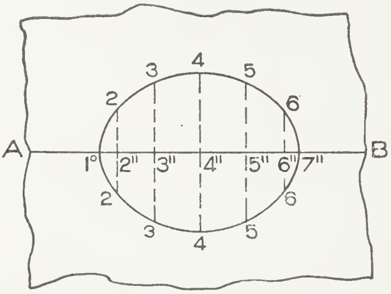


Fig. 136.

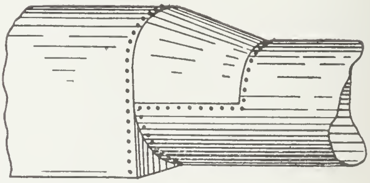


Fig. 137.

the intersections 1° to 4°, a line traced through these points will be the true section. In (Y) on any line as O P lay off the stretch-out of the true section as shown from 4°, 1°, 4°. As the gusset sheet only covers a portion equal to a half circle, add the distance 4° *e* equal to ½ of 7 times the thickness of the metal in use and

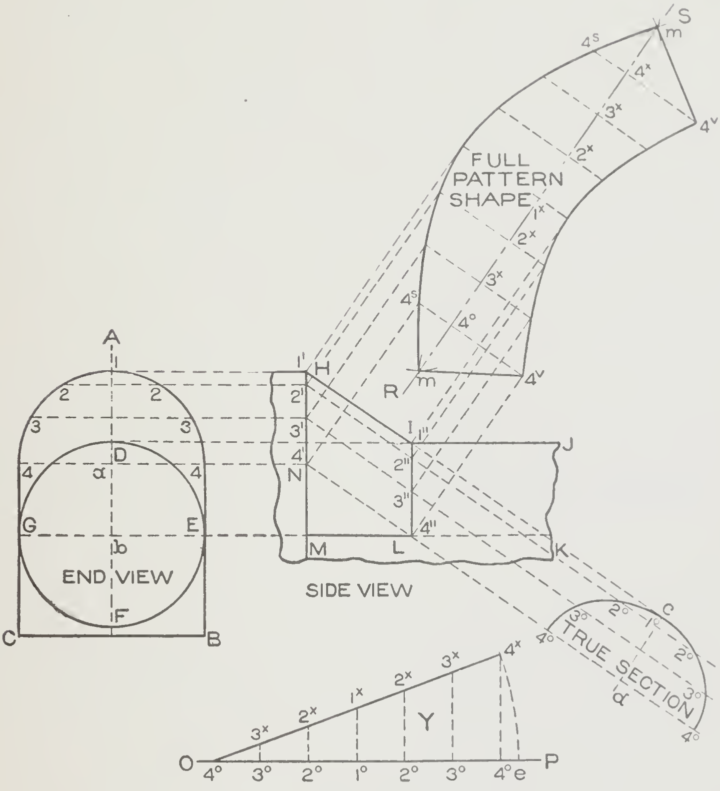


Fig. 138.

using 4° at the left, as center with 4° *e* as radius, describe the arc *e* 4^x, intersecting it at 4^x by the vertical line drawn from 4°. From O P erect vertical lines intersecting the line drawn from 4^x to 4 at 3^x, 2^x, 1^x, etc. 4° 4^x is the true stretchout, and should be placed on the line R S drawn at right angles to H I. Through the numbers on R S and at right angles draw the lines shown and intersect them by lines drawn from similarly numbered intersections on H N and I L at right angles to H I. Through points thus obtained trace a curved line 4^S, 4^S, and 4^V, 4^V. It now becomes necessary to add the triangular piece shown by L M N in side view, to the pattern which can be done as follows: Using L M in side view as radius and 4^V at either end of the pattern as centers, describe the arcs *m* and *n*; intersect them by arcs struck from 4^S and 4^S as centers, and M N in side view as radius. Then draw lines from 4^S to *m* to 4^V in the pattern on either side. The full pattern shape for the gusset sheet will then be shown by *m* 4^S 4^S *m* 4^V 4^V, to which laps must be allowed for riveting.

Fig. 139 shows a conical piece connecting two boilers with the flare of A such that the radial lines can be used in developing the pattern. In all such cases this method should be used in preference to that given in connection with Fig. 130. Thus in Fig. 139 the centers of the two boilers are on one line as shown by *a b*. While the pattern is developed the same as in flaring work, the method of allowing for the metal used is shown in Fig. 140.

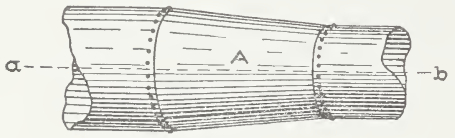


Fig. 139.

A B C D is the elevation of the conical piece, the half inside section being shown by 1 4 7 which is divided into equal spaces. 1 7 1 in (E) is the full stretchout of the inside section A 4 D in elevation, and 1 *e* is equal to 7 times the thickness of the metal used. The line 1 1′ is then obtained in the usual manner as are the various intersections 2′ 3′ 4′, etc. Now extend the lines A B and D C in elevation until they meet the center line *a b* at *a*. Then using *a c* and *a d* draw the arcs 1′ 7′ and 1″ 7″. From 1′ draw a radial line to *a*, intersecting the inner arc at 1″. Now set the dividers equal to the spaces on 1 1′ in (E) and starting from 1′ in the pattern step off 6 spaces and draw a line from 7′ to *a* intersecting the inner arc at 7″. 1′ 7′ 1″ 7″ shows the half pattern to which flanges must be allowed for riveting.

Fig. 141 shows a view of a scroll sign, generally made of heavy steel, heavy copper, or heavy brass. So far as the sign is concerned it is simply a matter of designing, but what shall be given attention here is the manner of obtaining the pattern and elevation of the scroll. As these scrolls are usually rolled up in form of a spiral, the method of drawing the spiral will first be shown.

Establish a center point as *a'* in Fig. 142, and with the desired radius describe the circle shown, which divide into a polygon of any number of sides, in this case being 6 sides or a hexagon. The more sides the polygon has, the nearer to a true spiral will the figure be. Therefore number the corners of the hexagon 1 to 5 and draw out each side indefinitely as 1 *a*, 2 *b*, 3 *c*, 4 *d*, 5 *e*, and 6 *f*. Now using 2 as center and 2 1 as radius, describe the arc 1 A; then using 3 as center and 3 A as radius, describe the arc

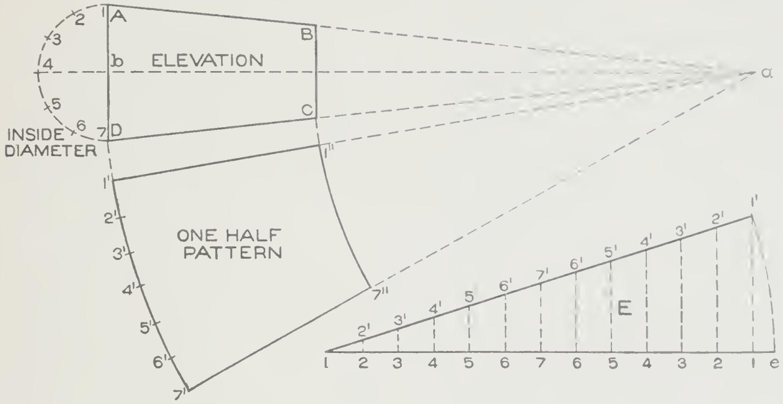


Fig. 140.

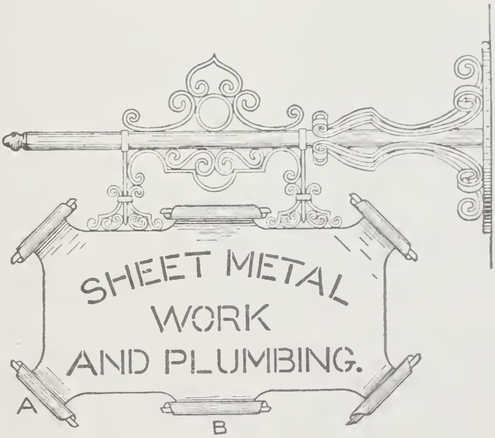


Fig. 141.

A B, and proceed in similar manner using as radii 4 B, 5 C, 6 D, and 1 E, until the part of the spiral shown has been drawn. Then using the same centers as before continue until the desired spiral is obtained, the following curves running parallel to those first drawn. The size of the polygon *a′*, determines the size of the spiral.

In Fig. 143 let A B C D represent the elevation of one corner of the flag sign shown in Fig. 141. In its proper position in Fig. 143 draw a section of the scroll through its center line in elevation as shown by *a* 17 to 1, which divide into equal spaces as shown from 1 to 17. Supposing the scroll is to be made of $\frac{1}{8}$ inch thick metal, and as the spiral makes two revolutions then multiply $\frac{1}{8}$ by 14, which would equal $1\frac{3}{4}$ inches. Then on E F in Fig 144 place the stretchout of the spiral in Fig. 143, as shown by similar numbers, to which add 17 E equal to 14 times the thickness of metal in use, and draw the arc E 17′ in the usual manner and obtain the true stretchout with the various intersections as shown. Through the elevation of the corner scroll in Fig. 143 draw the center line E F, upon which place the stretchout of 17′ E, Fig. 144, as shown by similar numbers on E F in Fig. 143. At right angles to E F, through 1′ and 17′, draw 17° 17° equal to A B and 1° 1° equal to the desired width of the scroll at that point. Then at pleasure draw the curve 1° 17° on either side, using the straight-

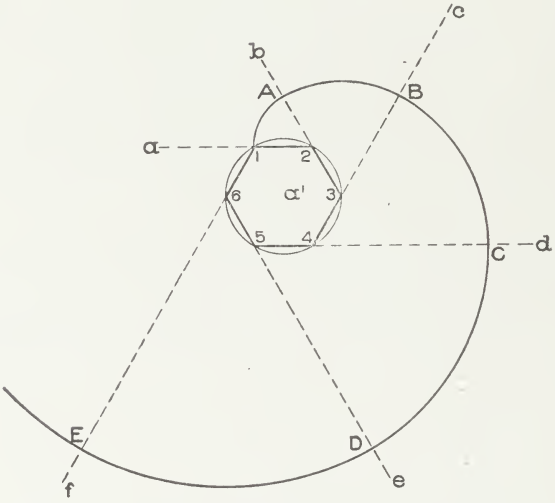


Fig. 142.

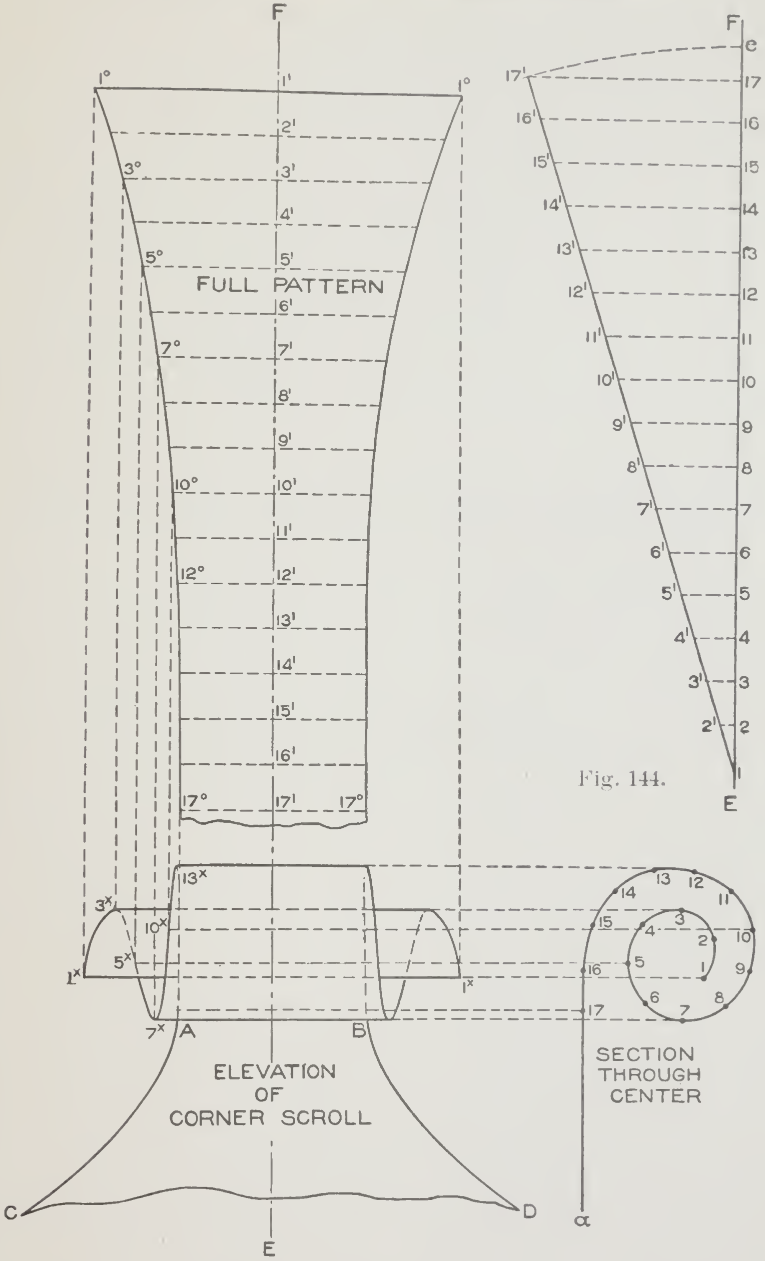


Fig. 144.

Fig. 143.

edge and bending it as required. Then will 1° 1° 17° 17° be the pattern for the scroll using heavy metal.

If it is desired to know how this scroll will look when rolled up, then at right angles to E F and through the intersections $1'$ to $17'$ draw lines intersecting the curves of the pattern 1°-17° on both sides. From these intersections, shown on one side only, drop lines intersecting similar numbered lines, drawn from the intersections in the profile of the scroll in section parallel to A B. To avoid a confusion of lines the points 1^x, 3^x, 5^x, 7^x, 10^x, 12^x, and 17^x have only been intersected. A line traced through points thus obtained as shown from 1^x to 17^x in elevation gives the projections at the ends of the scroll when rolled up.

EXAMINATION PAPER

Plates III to V, inclusive, constitute the Examination for this Instruction Paper. Plates IV and V are drawn by the student himself, and therefore no reproduced plates are sent. The date, student's name and address, and the plate number should be lettered on each plate in inclined Gothic capitals.

EXAMINATION PLATES.

PLATE III.

In this plate a set of pipe fittings is shown which should be drawn by the student carefully according to the measurements which will be given. If necessary, copy it before sending for examination and correction. This plate should be laid out the same size as previous plates and the border lines drawn as before. There are five fittings shown which will require six patterns. Reproduced plates of the patterns are not sent to the student; he should work out the problems for himself according to directions given. First draw in Plate III the rectangular section A B C D, making it $1\frac{3}{8} \times 2\frac{3}{8}$ inches, with the line A B $\frac{1}{2}$-inch below the margin line, and A B $\frac{1}{4}$-inch to the right of the margin line. Now with *a* in the section as center and with radius equal to $\frac{5}{8}$-inch draw the circle G H I J so that I will be tangent at D C and G I be central in the section. Then A B C D will be the true section on E F, while G H I J shows the section on the line K L. Draw E F one-quarter inch from the line B C, and at right angles to E F, draw F L equal to $2\frac{1}{2}$ inches and from L erect the line L K $1\frac{1}{4}$ inches or equal to the height of the circle I G in section. Draw the line K E. E F L K represents the side elevation of a transition piece whose base is rectangular and whose top is round, as shown in the section. The dotted lines in section and elevations show how the figure is divided into sections of scalene cones, necessary when developing the patterns. Now extend the line K L as K *b* and with radius equal to $1\frac{1}{2}$ inches draw the quadrant L M from *b* as center. Then using *b* again as center with *b* K as radius draw the outer arc K N, intersecting the horizontal line drawn from *b*. Draw the 3-piece elbow, whose section is shown by G H I J, as explained in connection with Fig. 42, Part I. Draw the center dotted line through the elbow and from *u* on the horizonal line *b* N extended lay off a distance equal to one inch as V^{O}. From V^{O} measure down on a vertical line a distance equal to $4\frac{3}{8}$ inches, to the point *w*. Using *w* as center with a radius equal to $1\frac{1}{4}$ inches describe the circle R S T. From *w* on the vertical line measure up a distance of $1\frac{3}{4}$ inches to x and through x draw the horizontal line O P, making x O and x P each $1\frac{1}{8}$ inches. From O and P drop vertical lines intersecting the circle at S and R. Draw M P and N O, forming the transition piece W, which connects round pipes with diameters equal to M N and P O respectively. X shows the side elevation of the collar connecting the main pipe R S T with the transition piece W.

A front elevation of the fittings should be drawn as follows: Draw the center dotted vertical line $c\,d$ $2\frac{5}{8}$ inches to the left of the margin line, and from the various intersections in the side elevation draw the dotted lines shown. Draw a broken view of the main pipe Y, $\frac{1}{4}$ inch from the margin line, and making both sides equal distance from the center line. Draw the intersection between the collar and main pipe as shown by i S^V i', drawing a curved line through these points. P^V O^V and M^V N^V represent the same diameters as P O and M N in side elevation. They are measured on either side of the center line $c\,d$ in front elevation. $e'\,f''\,h'\,f''$ is a front elevation through $e\,f\,h$ in side view, while L^1 K^1 is a true section on K L. A^1 B^1 C^1 D^1 is equal to A B C D in section. Draw dotted lines showing the transition from the rectangular section to round as given in the front elevation. When this has been done Plate III is completed.

We have now five fittings for which the patterns must be developed, and for which no reproduced plates are sent. The student should follow the rules given in previous problems.

PLATE IV.

The patterns for the transition piece U and the 3-pieced elbow V constitute Plate IV.

To obtain the patterns for the transition piece U, use the description given in connection with Figs. 14 and 15, but lay off the diagram of triangles similar to the directions given in connection with Figs. 78 to 81. The patterns for the 3-pieced elbow V should be developed as described in connection with Figs. 46 and 47. These two problems U and V should be carefully laid out on similar sized plate as previously used. Care should be taken not to allow any of the patterns to come within $\frac{1}{4}$ inch of the margin line and place the patterns in such positions to make a neat appearance.

PLATE V.

The patterns of W and X, together with the pattern for the opening in Y, constitute this plate.

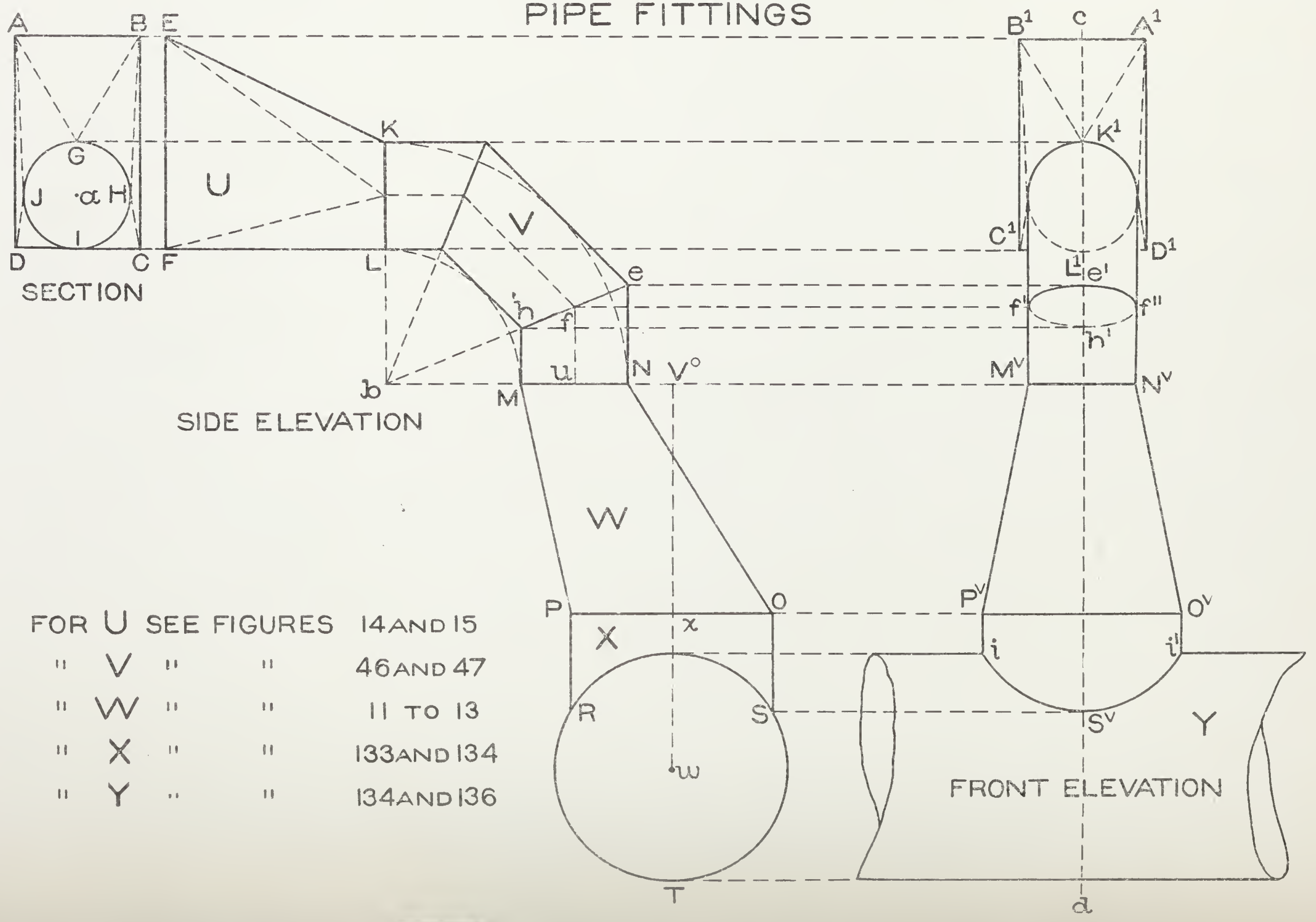
To obtain the pattern for the transition piece W, use the rule described in connection with Figs. 11 to 13, and for the pattern X use rules described in connection with Figs. 133 and 134. In this case we make no allowance for the thickness of the metal as shown in those figures, but assume that we are using ordinary gauge metal. For the pattern for the opening in the main pipe Y use rules given in connection with Figs. 134 and 136, and also omit allowing for heavy material.

EXAMINATION PLATES.

Plates III to V, inclusive, constitute the Examination for this Instruction Paper. As above mentioned, Plates IV and V are drawn by the student himself, and therefore no reproduced plates are sent. The date, student's name and address, and the plate number should be lettered on each plate in inclined Gothic capitals.

PLATE III

PIPE FITTINGS



FOR U SEE FIGURES 14 AND 15
" V " " 46 AND 47
" W " " 11 TO 13
" X " " 133 AND 134
" Y " " 134 AND 136

PRACTICAL PROBLEMS IN MENSURATION FOR SHEET METAL WORKERS.

A square tank, Fig. 1, is required whose capacity should be 200 gallons, the sides *b a* and *a c* each to be 30 inches; how high must *c d* be, so that the tank will hold the desired quantity?

Suppose the height *c d* is to be $51\frac{1}{3}$ inches, and the tank is to have similar capacity, and one side *c a* is to be 20 inches wide, how long must the alternate side *a b* be, so that the tank will hold 200 gallons?

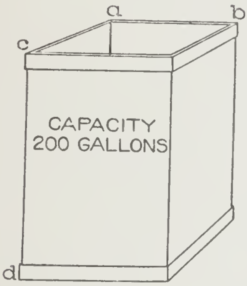


Fig. 1.

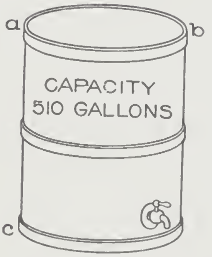


Fig. 2.

A round tank, Fig. 2, is to be constructed whose capacity should equal 510 gallons, and be 5 feet high from *c* to *a*; what must its diameter *a b* be, so as to hold the desired capacity?

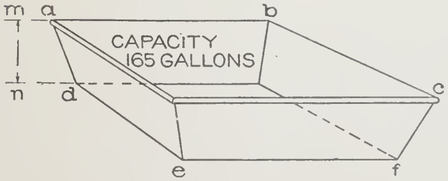


Fig. 3.

Suppose the diameter of the tank is to be 50 inches as *a b*; what must its height *a c* be, so that the tank will hold 510 gallons?

A large drip pan, Fig. 3, is to be constructed whose capacity should be 165 gallons, and whose top measurements *a b* and *b c* are 60 × 40 inches respectively, and bottom measurements *d e* and *e f* 34 × 54 inches respectively; what must its height *m n* be, so as to hold the desired volume?

A round tapering measure, Fig. 4, is to be constructed whose volume will equal 42 quarts; its bottom diameter *a b* is to be 14 inches, its top diameter *c d* 18 inches; what must its height *e f* be to hold the desired quantity?

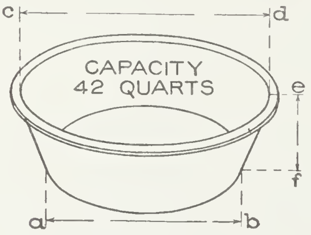


Fig. 4.

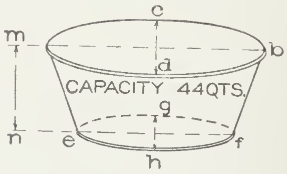


Fig. 5.

An elliptical tapering tank, Fig. 5, is to be constructed whose major axis *m b* is 24 inches, and minor axis *c d* 14 inches at the top, while at the bottom the major axis *e f* is 20 inches, and minor axis *g h* 10 inches; the capacity of the tank should equal 44 quarts; what must the height *m n* be, so that the tank will hold the desired amount?

A tank, Fig. 6, is to be constructed with semicircular ends whose capacity should equal 30 gallons; the length *a b* to be 20 inches, and the diameters of *c* and *d* to be each 10 inches; what must the height *e f* be, so that the tank will hold the desired quantity?

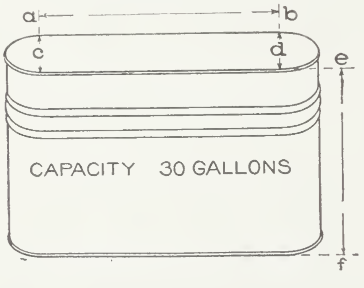


Fig. 6.

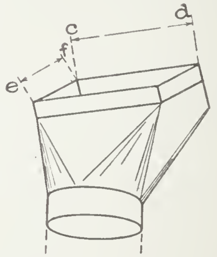


Fig. 7.

Suppose the height *e f* is to be 24 inches, the diameters *c* and *d* each 11 inches; what must the length of *a b* be, so that the tank will hold 30 gallons?

In Fig. 7 is shown a fitting used in ventilation piping; the diameter $a\,b$ is $11\frac{1}{4}$ inches and it is desired that the oblong pipe on the opposite end shall have an area similar to the round pipe $a\,b$; if $e\,f$ must be 5 inches, what must $c\,d$ be so that both areas are alike?

Suppose the pipe is to be square in place of oblong, what must the length of each side be, so that both ends have similar area?

In Fig. 8, $a\,b$ is 40 inches in diameter; and each one of the branches c, d, and e are to have equal diameters, what must the diameter of the branches be, so that the combined area of c, d, and e will equal the area of $a\,b$?

If c is 10 inches in diameter, d 12 inches, and e 8 inches, what must be the diameter of $a\,b$, to have the combined area of the branches?

Fig. 9 shows a transition piece from a round pipe a to an elliptical pipe b, both sections to have similar area; if the round pipe is 24 inches in diameter, and the major axis of the elliptical pipe must be 32 inches, what must the minor axis of b be so that the area at b will equal the area of a?

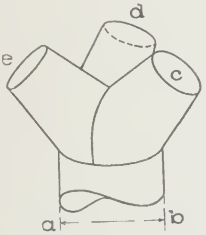


Fig. 8.

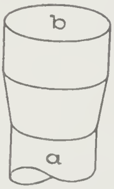


Fig. 9.

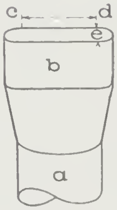


Fig. 10.

If the minor axis of b is to be 16 inches and the major axis 35 inches, what must the diameter of a be, so that both sections will have similar area?

In Fig. 10, a is 20 inches in diameter and forms a transition to an oblong pipe with semicircular end; the semicircular ends are to be 10 inches in diameter; what must the length of $c\,d$ be, so that the area of b will be equal to the area of a?

If the pipe b measured 40 × 11 inches, having semicircular ends, what must the diameter of a be, so that both sections are equal in area?

If a is 20 inches in diameter and the upper section was to be rectangular in shape, 8 inches wide, what would the length of the upper section be?

Suppose the upper section *b* was desired to be square, what must the length of each side be, to have an area similar to *a*?

In Fig. 11 is shown the illustration of an ordinary steel square, and the method is given of obtaining accurate diameters of pipes, round or square, without any computation whatever, the rule being based on the geometrical principle that the square of the hypothenuse of a right angle triangle is equal to the sum of the squares of its base and altitude. To illustrate the rule, Fig. 12 has been prepared. Let A represent a round or square pipe, 20 inches across, and B a round or square pipe 12 inches across; it is desired to take a branch from the main so that the two branches B and C will equal the area of the main A. What must the size of C be?

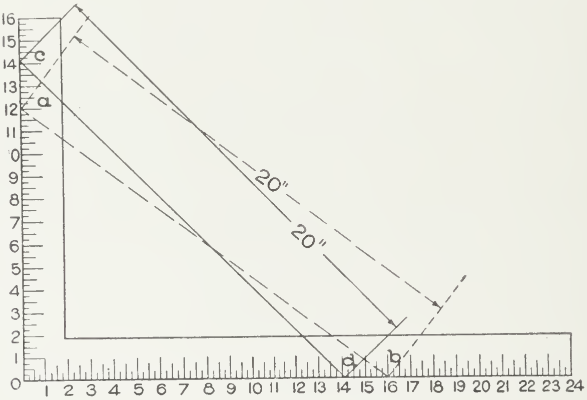


Fig. 11.

The size of C is found by simply taking a rule 20 inches long and placing one end on the arm of the square in Fig. 11, on the number 12, when the opposite end of the rule will touch the number 16. Then 16 is the required size of the branch C in Fig. 12. We can prove this by computation which, however, is not necessary in practice. The area of a 20-inch round pipe equals 314.16 in.; area of 12-in. pipe = 113.098 in.; area of 16-in. pipe = 201.062 in.; and 113.098 in. + 201.062 in. = 314.160 in. The area of a 20-in. square pipe = 400 in.; area of 12-in. square pipe = 144 in.; area of 16-in. square pipe = 256 in.; and 256 in. + 144 in. = 400 in.

Suppose any two branches are given as B and C in Fig. 12, what must the size of A be so that its area will have the combined area of the two branches?

Simply set the rule on the numbers 12 and 16 on the two arms of the square respectively, and the length from *a* to *b* in Fig. 11 will measure 20 inches.

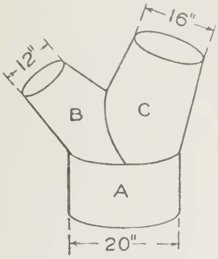
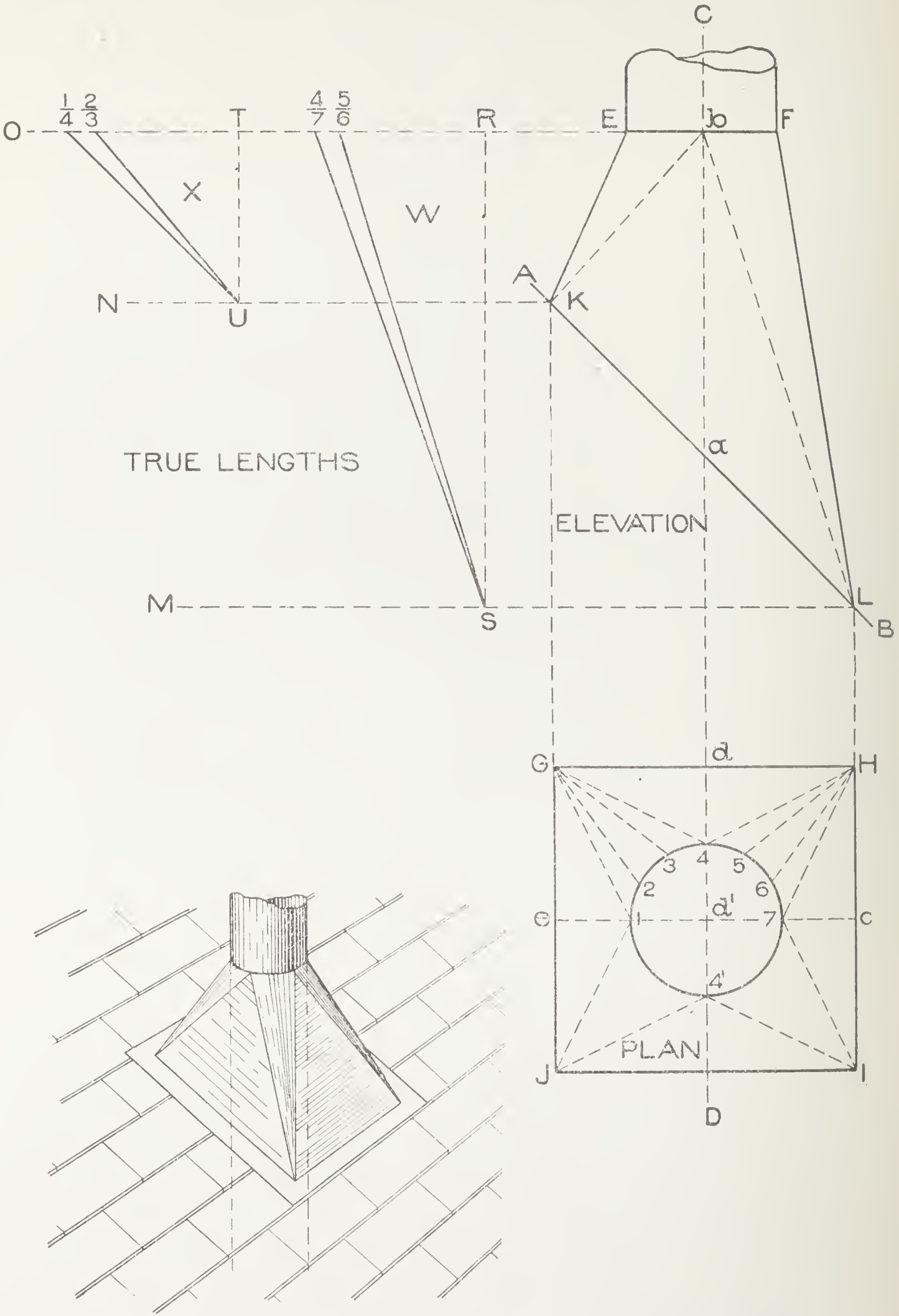


Fig. 12.

If A, Fig. 12, were given, and two branches were required, so that B and C were both of equal size, then simply set the rule 20 inches long, on both arms of the square so that the distance from O to *c* and O to *d* would be equal, as shown in Fig. 11, which would be found to measure $14\frac{1}{7}$ in. plus a least trifle. This rule can be used to advantage for any size round or square pipe in blower, blast, heat, and ventilating piping, saving time and trouble in computation. Where no square is at hand, one can be drawn on paper and used for work of this kind.



PATTERN OF TAPERING FLANGE FOR CYLINDER, DEVELOPED BY TRIANGULATION.

TINSMITHING.

An important part of the technical education of those connected with tinsmiths' work is a knowledge of laying out patterns. When making the various forms of tinware, or, as they are commonly called, housefurnishing goods, the greatest care must be taken in developing the patterns, for if a mistake of but one point is made, the pattern will be useless. There are general geometrical principles which are applied to this work which, when thoroughly understood, make that part plain and simple, which would otherwise appear intricate. These principles enable the student to lay out different patterns for various pieces of tinware where the methods of construction are similar.

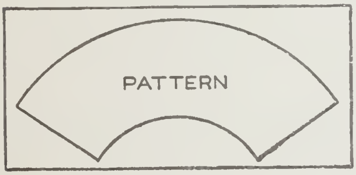


Fig. 1.

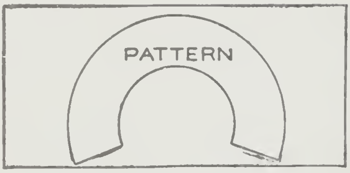


Fig. 2.

Construction. Before laying out the pattern for any piece of tinware, the method of construction should be known. Knowing this, the first thought should be: Can the pattern be developed and cut from one piece of metal to advantage, as shown in Fig. 1, or will it cut to waste, as shown in Fig. 2 ? Will the articles have soldered, grooved or riveted seams, as shown respectively by A, B and C, in Fig. 3 ? Also, will the edges be wired or have hem edges at the top, as shown respectively by A and B, in Fig. 4 ? Sometimes the pattern can be laid out in such a way that the article may be made up of two or more pieces, so that the patterns may be laid in one another, as shown in Fig. 5, thereby saving material. This is a plan that should always be followed if possible.

When the patterns are developed, tin plate should be obtained of such size as to have as little waste as possible.

By means of the table on pages 45–47 tin plate may be ordered which will cut to advantage, for there is nothing worse in a tinshop than to see a lot of waste plate under the benches, whereas a little foresight in ordering stock would have saved material.

Capacity of Vessels. Sometimes the tinsmith is required to make a piece of tinware which will hold a given quantity of liquid. The methods of finding the dimensions are given in Arithmetic and Mensuration, which subjects should be reviewed before beginning this work.

Shop Tools. The most important hand tools required by the tinsmith are: hammer, shears, mallet, scratch awl, dividers and soldering coppers. The other tinsmith tools and machines will be explained as we proceed.

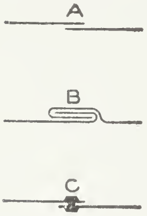


Fig. 3.

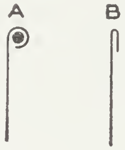


Fig. 4.

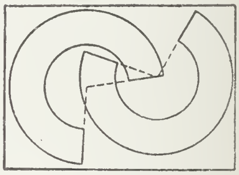


Fig. 5.

Various Methods of Obtaining Patterns. The pattern drafting for this course is divided into two classes:

1. Patterns which are developed by means of parallel lines.
2. Patterns which are developed by means of radial lines.

The principles which follow are fundamental in the art of pattern cutting and their application is universal in tinsmiths' work.

INTERSECTIONS AND DEVELOPMENTS.

The laying out of patterns in tinsmiths' work belongs to that department of descriptive geometry, known as development of surfaces, which means the laying out flat of the surfaces of the solids, the flat surfaces in this case being the tinplate. In Fig. 6 is shown one of the most simple forms to be developed by parallel lines, that of an octagonal prism. This problem explains certain fixed rules to be observed in the development of all parallel forms, which are as follows:

1. There must be a *plan*, *elevation* or other view of the article to be made, showing the line of joint or intersection, and

in line with which must be drawn a section or profile of the article. Thus, ABCD shows the view of the article, AL the line of joint or intersection, and E the profile or section of the article.

2. The *Profile* or section (if curved) must be divided into equal spaces (the more spaces employed the more accurate will be the pattern), from which lines are drawn parallel to the lines of the article intersecting the line of joint or intersection. Thus from the corners numbered 1 to 8 in the profile E, lines are drawn parallel to the line of the article, intersecting the line of joint AL from 1″ to 8″. In Fig. 7, where the section A is curved, this is divided into equal spaces.

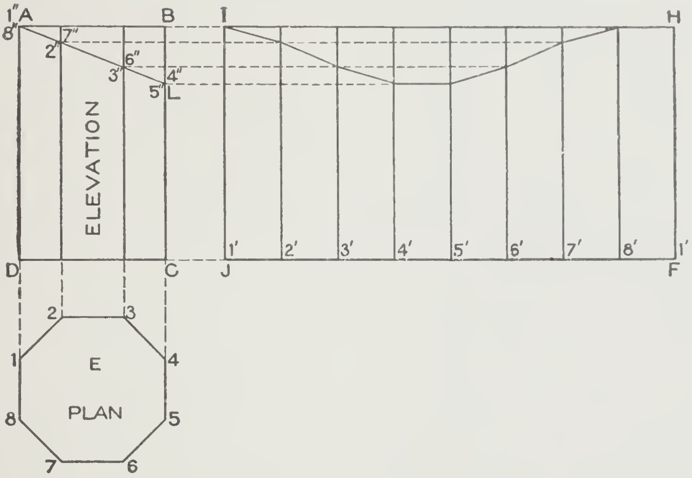


Fig. 6.

3. A *stretchout* line (showing the amount of material the article will require) is next drawn at right angles to the line of the article, upon which is placed each space contained in the section or profile. Thus JF, in Fig. 6, is the stretchout line, which contains the true amount required to enclose the profile E.

4. At right angles to the stretchout line, and from the intersections thereon, draw lines called the *measuring lines*. Thus, from the intersections 1′ to 8′ on JF lines are drawn at right angles to the stretchout line JF, which are called measuring lines.

5. From the intersections on the line of joint draw lines intersecting similarly numbered measuring lines, which will result in the pattern shape. Thus lines drawn from the intersections on

the line AL at right angles to BC intersect similarly numbered measuring lines as shown. Then JIHF will be the development for an octagonal prism intersected by the line AL in elevation.

This simple problem shows the fundamental principles in all parallel-line developments. What we have just done is similar to taking the prism and rolling it out on a flat surface. Let the student imagine the prism before him with the corners blackened and starting with corner 1 turn the prism on a sheet of white paper until the point 1 is again reached, when the result will correspond to the development shown. Bearing these simple rules in mind, the student should have no difficulty in laying out or developing the forms which will follow.

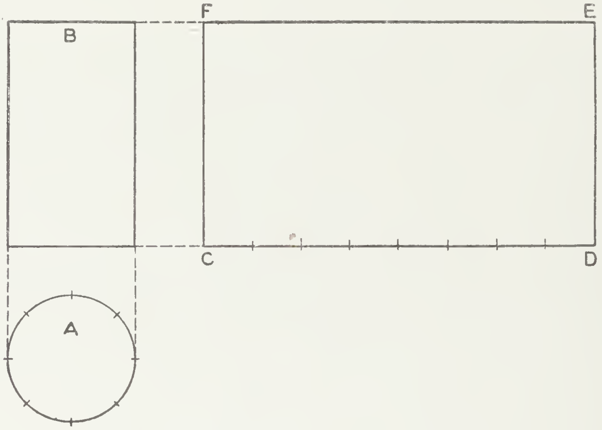


Fig. 7.

Fig. 7 shows the development of a cylinder, and also shows the principles which are applied in spacing circular sections or profiles, as explained for parallel developments. A shows the profile or section, B the elevation, and CD the stretchout line or the amount of material required to go around the circle. By drawing the measuring lines CF and DE and connecting them by the line FE, we obtain CDEF, which is the development of the cylinder.

Fig. 8 shows how to obtain the development of the surfaces of an intersected hexagonal prism, the angle of intersection being 45°. First draw the elevation ABCD and the section E in its proper position below. Number the corners in the section 1, 2 and 3, as shown, from which erect perpendicular lines intersecting the plane AB, as shown by 1°, 2° and 3°. Bisect the lines 1—1 and 3—3 in plan obtaining the points F and H respectively, and draw the line FH. This line will be used to obtain dimensions with which to construct the developed surface on the plane AB. At right angles to AB and from the intersections 1°, 2° and 3° draw lines as shown. Parallel to AB draw the line F^v H^v. Now, measuring in each instance from the line FH in E, take the distances to 1, 2 and 3, and place them on similarly numbered lines drawn from the plane AB, measuring in each instance from the

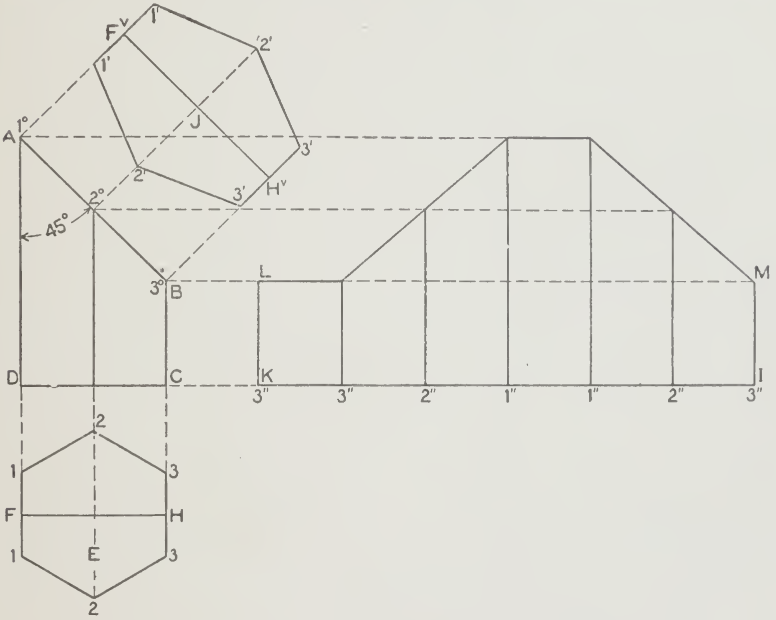


Fig. 8.

line $F^v H^v$ on either side, thus obtaining the points 1′, 2′ and 3′. Connect these points by lines as shown; then J will be the true development or section on AB.

For the development of the prism, draw the stretchout line KI at right angles to AD, upon which place the stretchout of the section E, as shown by similar numbered intersections on KI. From these intersections, at right angles to KI, draw the measuring lines shown, which intersect with lines drawn from similar numbered intersections on the plane AB, at right angles to BC. Through the intersections thus obtained, draw the lines from L to

M. Then KLMI will be the pattern or development of the intersected prism.

Fig. 9 shows the development of an intersected cylinder. A is the elevation and B the profile or plan. As each half of the development will be symmetrical, divide the profile B into a number of equal parts, numbering each half from 1 to 5, as shown. From these points perpendicular lines are erected, intersecting the plane 1^v — 5^v at 1^v, 2^v, 3^v, 4^v and 5^v. A stretchout is now made of the profile B and placed on the horizontal stretchout line CD, the points being shown by 5′, 4′, 3′, 2′, 1′, 2″, 3″, 4″ and 5″. From these points measuring lines are erected and intersected by similar numbered lines drawn from the plane 1^v — 5^v at right angles to the line of the cylinder. A line traced through points thus obtained will be the development of the intersected cylinder. In this case the butting edge or joint line of the cylinder is on its shortest side. If the butting edge were desired on its longest side, it would be necessary to change only the figures on the stretchout line CD, making 1′ start at 5′ and end at 5″.

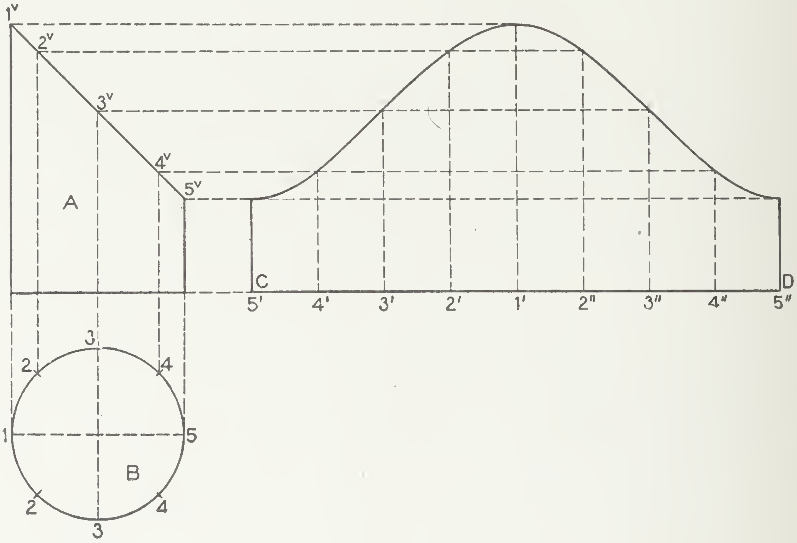


Fig. 9.

Where two prisms intersect each other, as shown in Fig. 10, it is necessary to find the points of intersection before the surfaces can be developed. Thus we have two unequal quadrangular prisms intersecting diagonally at right angles to each other. We first draw the section of the horizontal prisms as shown by B in the end view, from which the side view A is projected as shown. From the corner T in the section B erect the perpendicular line TC, and above in its proper position draw the section D of the vertical prism, and number the corners 1, 2, 3 and 4. From the corners 1 and 3 drop vertical lines intersecting the profile B at 1′ and 3′, T representing the points 2′ and 4′ obtained from 2 and 4 in D. From the points 1′ and 3′ in B, draw a horizontal line through the side view, and locate the center of the vertical prism as 3″, from which erect the perpendicular line 3″ — 1. Now take a duplicate of the section D and place it as shown by F, allowing it to make a quarter turn (90°); in other words, if we view the vertical prism from the end view, the point 1 in section D faces the left, while if we stood on the right side of the end view the point 1 would point ahead in the direction of the arrow. The side view therefore represents a view standing to the right of the end view, and therefore the section F makes a quarter turn, bringing the corner 1 toward the top. From points 2 and 4 in section F drop vertical lines intersecting the line drawn from the corner 2′ — 4′ in B, thus obtaining the intersections 2″ — 4″ in the side view. Draw a line from 4″ to 3″ to 2″, which represents the intersection between the two prisms.

To develop the vertical prism, draw the horizontal stretchout line HI, and upon it place the stretchout of the profile D as shown by similar figures on HI. Draw the measuring lines from the points 1, 2, 3, 4, 1, at right angles to HI, which intersects with lines drawn at right angles to the line of the vertical prism from intersections having similar numbers on B. A line traced through the points thus obtained, as shown by HILJ will be the development of the vertical prism. The development of the horizontal prism with the opening cut into it to admit the joining of the vertical prism is shown in Fig. 11, and is drawn as follows: Draw any vertical line O^v P^v, and on this line place the stretchout of the upper half of section B in Fig. 10, as shown by similar letters and figures in Fig. 11. From these points at right angles to O^v P^v draw lines equal in length to the side view in Fig. 10. Draw a line from U to T in Fig. 11. Now, measuring from the line RS in side view in Fig. 10, take the various distances to points of in-

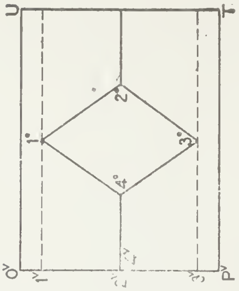

U
T
1°
2°
3°
4°
O^v
1^v
2^v
3^v
P^v

Fig. 11.

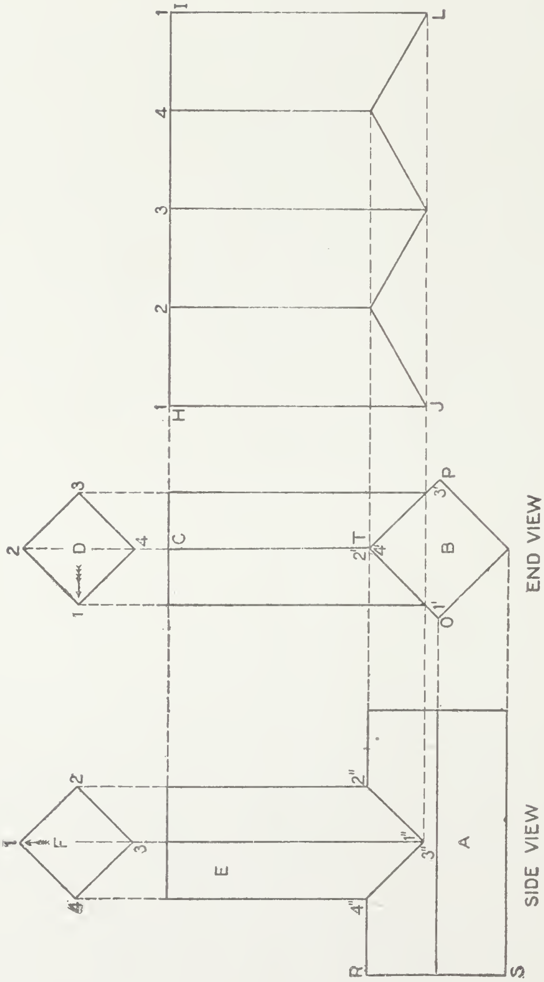

I
L
1
4
3
2
1
H
J
3
2
D
4
1
C
T
2'
4'
B
3'
P
1'
O
END VIEW
2
1
F
3
2"
1"
3"
4"
4
E
A
R
S
SIDE VIEW

Fig. 10.

tersections 4″, 3″, 1″ and 2″, and place them in Fig. 11 on lines having similar numbers, measuring from the line $O^v P^v$, thus resulting in the intersections 1^v, 2^v, 3^v and 4^v. Connecting these points by lines as shown, then O^vUTP^v will be the half development of the top of the horizontal prism. The bottom half will he similar without the opening.

Having described the principles relating to parallel forms, the next subject will be the principles relating to tapering forms. These forms include only the solid figures that have for a base the circle, or any of the regular polygons, also figures of unequal sides which can be inscribed in a circle, the lines drawn from the corners of which terminate in an apex, directly over the center of the base. The forms with which the tinsmith has to deal are more frequently frustums of these figures, and the method used in developing these surfaces is simply to develop the surface of the entire cone or pyramid, and then by simple measurements cut off part of the figure, leaving the desired frustum. Thus in the well-known forms of the dipper, coffee pot, colander, strainer, wash bowl, bucket, funnel, measure, pan, etc., we have the frustums of cones above referred to. In speaking here of metal plate articles as portions of cones, it must be remembered that all patterns are of surfaces, and as we are dealing with tinplate, these patterns when formed are not solids, but merely shells. In works upon Solid Geometry the right cone is defined as a solid with a circular base, generated by the revolution of a right-angle triangle about its vertical side called the axis.

This is more clearly shown in Fig. 12, in which is shown a right cone, which contains the principles applicable to all frustums of pyramids and cones. ABC represents the elevation of the cone; the horizontal section on the line BC being shown by GDEF, which is spaced into a number of equal parts, as shown by the small figures 1 to 12. As the center or apex of the cone is directly over the center a of the circle, then the length of each of the lines drawn from the small figures 1 to 12 to the center a will be equal both in plan and elevation. Therefore to obtain the envelope or development, use AB or AC as radius, and with A in Fig. 13 as center, describe the arc 1 – 1′. From 1 draw a line to A and starting from the point 1, set off on the arc 1 – 1′ the stretchout or number of spaces contained in the circle DEFG in Fig. 12, as shown by similar figures in Fig. 13. From 1′ draw a line to A. Then A – 1 – 7 – 1′ will be the development of the right cone of Fig. 12.

Suppose that a frustum of the cone is desired as shown by HICB, Fig. 12; then the opening at the top will be equal to the small circle in plan, and the radius for the pattern will be equal to AI. Now using A in Fig. 13 as a center with AI as radius, describe the arc HI, intersecting the lines 1A and A1′ at H and I respectively. Then H – I – 1′ – 7 – 1 will be the development for the frustum of the cone.

When a right cone is cut by a plane passed other than parallel to its base, the method of development is somewhat different. This is explained in connection with Fig. 14, in which A is the right cone, intersected by the plane represented by the line DE. B represents the plan of the base of the cone, whose circumference is divided into equal spaces. As the intersection of both halves of the cone are symmetrical, it will be necessary to divide only half of plan B as shown by the small figures 1 to 7. From these points, erect lines parallel to the axis of the cone, intersecting the base line of the cone. From these points draw lines to apex F, intersecting the line DE as shown. From the intersections thus obtained on the line DE and at right angles to the axis, draw lines as shown, intersecting the side of the cone FE. Now using F as center and FH as a radius, describe the arc 7 – 7′. From 7 draw a line to F, and

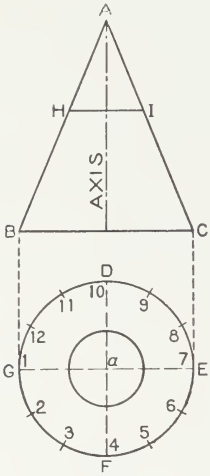


Fig. 12.

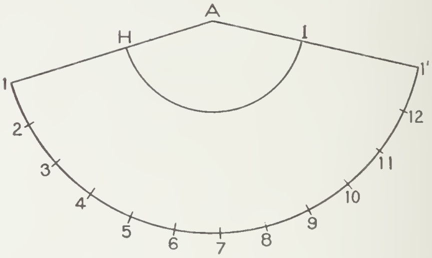


Fig. 13.

starting from the point 7 set off on the arc 7 – 7′, the stretchout of the circle B as shown by the small figures 7 – 1 – 7′. From these points draw radial lines to the center point F, and intersect them by arcs struck from the center F, with radii equal to similarly numbered intersections on the side FH, and partly shown by points $7^v - 1^v - 7^\circ$. Trace a line through the points of intersections thus obtained; then $7^\circ - 7^v - 7 - 7'$ will be the desired development.

These same principles are applicable no matter at what angle the cone is intersected. For the section on the line DE, see the explanation in Mechanical Drawing Part III.

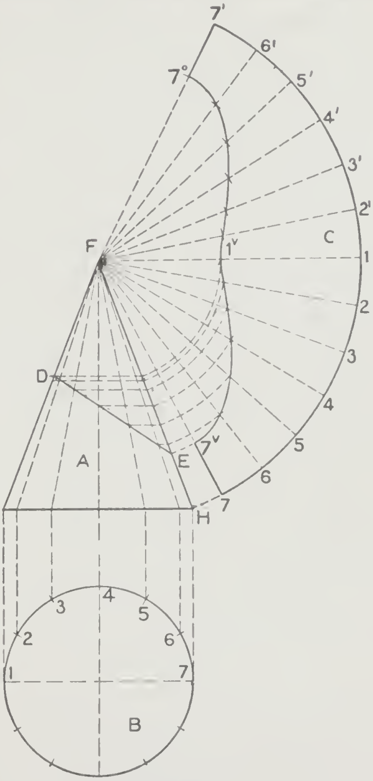


Fig. 14.

Fig. 15 shows the principles applicable to the developments of pyramids having a base of any shape. In this case, we have a square pyramid, intersected by the line DE. First draw the elevation of the pyramid as shown by ABC and in its proper position the plan view as shown by 1, 2, 3, 4. Draw the two diagonal lines 1 – 3 and 2 – 4 intersecting each other at A′. The length of the line AC represents the true length on A′*e*, but is not the correct radius with which to strike the development.

A true length must be obtained on the line A′4 as follows: At right angles to 3 – 4 from the center A′ draw the line A′E′ and using A′ as center and A′4 as radius, describe the arc 4E′ intersecting A′E′ at E′. From E′ erect the perpendicular line $E'1^v$ intersecting the base line BC extended at 1^v. From 1^v draw a straight line to A, which will be the true length on A′4 and the radius with which to strike the development. (See also Part III, Mechanical Drawing) Now with A as center and $A-1^v$ as radius, describe the arc $1^v-3^v-1^v$. Starting from 1^v set off the stretchout of $1-2-3-4-1$ in plan, as shown by 1^v–2^v–3^v–4^v–1^v on the arc 1^v-1^v (1^v-2^v being equal to $1-2$, etc.), and from these points draw lines to the apex A and connect points by straight lines as shown from 1^v to 2^v, 2^v to 3^v, 3^v to 4^v and 4^v to 1^v. Then $A1^v3^v1^v$ will be the development of the square pyramid.

To obtain the cut, in the development of the intersected plane

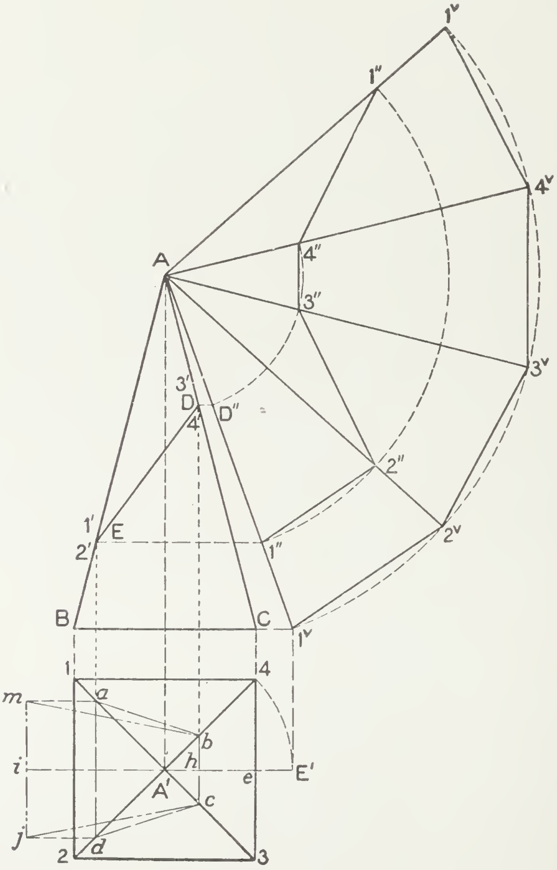


Fig. 15.

DE, which represents respectively the points 3^1-4^1 and 1^1-2^1, draw at right angles to the center line, the lines $D-D''$ and E–$1''$, intersecting the true length $A1^v$ at D'' and $1''$. Using A as center and radii equal to $A-D''$ and $A-1''$ intersect similarly numbered radial lines in the development. Connect these points as shown from 1″ to 2″, 2″ to 3″, 3″ to 4″ and 4″ to 1″. Then $1'' - 1^v - 3^v - 1^v - 1'' - 3''$ will be the development of the intersected square pyramid.

To draw DE in plan drop perpendiculars from D and E intersecting the diagonal lines in plan at *b c* and *d a*. Connect lines as shown at *a*, *b*, *c* and *d*. To obtain the true section of the plane DE, take the length of DE and place it, as shown in plan from *h* to *i*; through *i* draw the vertical line *jm* which is intersected by horizontal lines drawn from points *a* and *d*. Draw a line from *b* to *m* and *c* to *j* which will be the desired section.

These problems just described should be thoroughly studied and practiced on paper, until every step is well understood.

Practical Workshop Problems will now be considered, and the student who thoroughly understands the principles explained in the foregoing problems, will be able to develop the patterns with greater ease and in less time than is required by the student, who pays little attention to the principles, but simply proceeds to develop the patterns by blindly following directions. A thorough knowledge of the principles renders the student independent as far as pattern problems are concerned, as he can apply them to new work.

Short Rules. There are various short rules, which, while not geometrically accurate, are sufficiently so for all practical purposes and will be introduced as we proceed. In developing patterns for any given article, the problem should be gone over carefully, locating the joints or seams, so that it can be seen, we might say in our minds' eye; by doing this a shorter rule may be employed, thus saving time and expense. The student who pays attention to these smaller details will succeed as a pattern draftsman.

Allowance for Seaming and Wiring. As we are dealing with tin plate only, we assume this to have no thickness, and therefore make no allowance for the shrinkage of the metal, when bending in the machine folder or brake.

The amount of the material to be added to the pattern for wiring will vary according to the thickness of the metal. A safe and practical plan is to use a small strip of thin metal about $\frac{1}{4}$ inch wide and curl this around the wire which is to be used as shown in Fig. 16. This will give the true amount of material required, whether the wire is to be laid in by hand or by means of the wiring machine. First bend off with plyers a sharp corner as shown at *a*, place the wire in the corner and turn A snugly around the wire as shown at B. The amount of A, or the allowance to be added to the height of the pattern is thus obtained. The vertical joint in tinware is usually a lock seam as shown in Fig. 17. Three times the width of the lock *a* must be added to the pattern. In other words, the end *b* has a single edge as *d*, while the other end *c* has a double edge as shown at *a* and *e*; the two ends of the body joining at *f*.

In allowing these edges for the pattern, some workmen prefer to add a single edge on one side of the pattern, and a double edge on the other, while others prefer to allow one-half of the amount required on either side of the pattern. Where the bottom of any piece of tinware is to be joined to the body, it is generally double seamed as is shown in Fig. 18, where the two operations are clearly shown by A and B whether the seaming is done by hand or machine, while the lock seam in Fig. 17, is done on the groover.

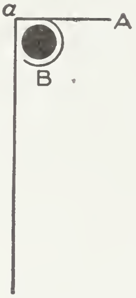


Fig. 16.

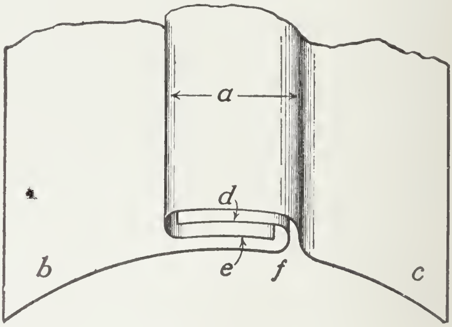


Fig. 17.

Notching the Patterns. Another important point is the notching of the edges of the patterns for seaming and wiring; special attention should be given to this. The notches should be made in such a manner that when the article is rolled up and the wire encased or the seams grooved, the ends of the wire or seam allowance will fit snugly together and make a neat appearance. When an article is made and the notches have not been cut properly, the wire, or uneven lines, will show at the ends of the seam. Fig. 19 shows how the allowance for wire or locks should be cut. A shows the pattern to which an allowance has been made for wire at B and for seaming to the bottom at C. In this case a single edge D has been allowed at one end of the pattern and a double edge of the other as shown at E. Then, using this method of allowance for seaming, notch the allowance for wire B and seam C on a line drawn through the solid lines in the pattern as shown by *aa* and *bb*. The notches of the allowance D and E should be cut at a small angle, as shown.

Transferring Patterns. After the pattern has been developed on manilla paper, which is generally used in the shop, it is placed on the tin plate and a few weights laid on top of the paper; then with a sharp scratch awl or prick punch and hammer, slight prick-punch marks are made, larger dots indicating a bend. The paper is then removed and lines scribed on the plate, using the scratch awl for marking the straight lines, and a lead pencil for the curved lines. After laps are added as required, it is ready to be cut out with the shears.

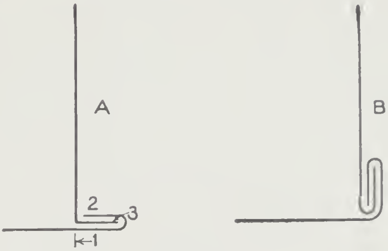


Fig. 18.

PRACTICAL PROBLEMS.

In presenting the twelve problems which follow, particular attention has been given to those problems which arise in shop practice. These problems should be practiced on cheap manilla paper, scaling them to the most convenient size, and then proving them by cutting the patterns from thin card board, and bending or forming up the models. This will prove both instructive and interesting.

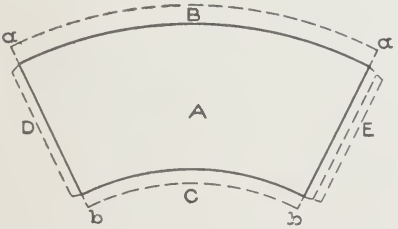


Fig. 19.

Pail. The first piece of tinware for which the pattern will be developed is that known as the flaring bucket, or pail, shown in Fig. 20. First draw the center line AB, Fig. 21, upon which place the height of the pail, as shown by CD. On either side of the center line place the half diameters CE of the top and DF of the bottom. Then EFFE will be the elevation of the pail. Extend the lines EF until they meet the center line at B, which will be the center point with which to describe the pattern. Now, with C as center and CE as radius, describe the semi-circle EAE, and divide it into equal spaces, as shown. This semi-circle will represent the half section of the top of the pail.

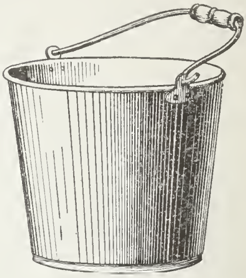


Fig. 20.

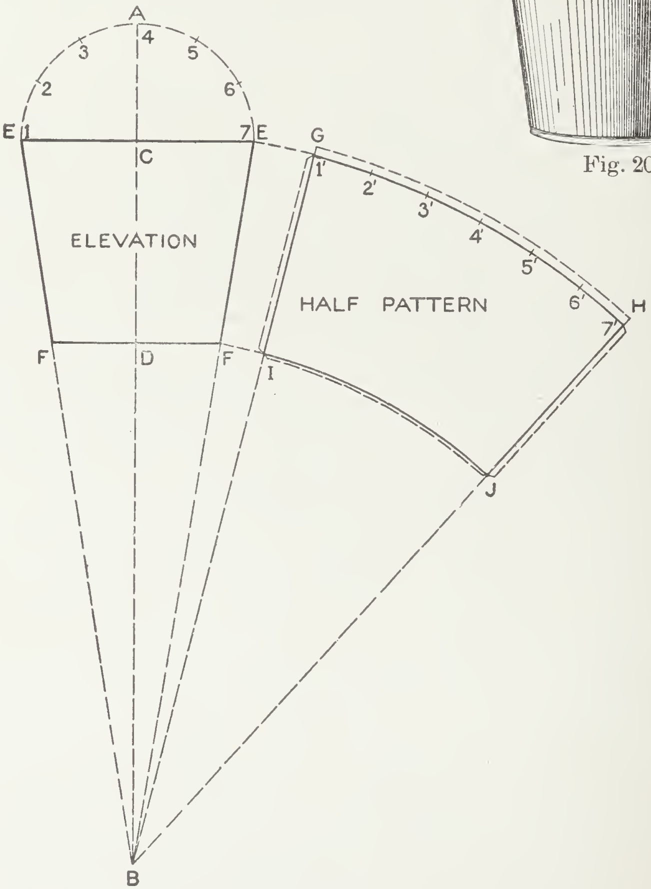


Fig. 21.

For the pattern proceed as follows: With B as center and radii equal to BF and BE, describe the arcs GH and IJ. Draw a line from G to B. Starting from the point G lay off on the arc GH, the stretchout of the semi-circle EAE, as shown by similar figures on GH. From H draw a line to B, intersecting the arc IJ at J. Then GHJI will be the half pattern for the pail, to which laps must be added for seaming and wiring as shown by the dotted lines.

Funnel and Spout. In Fig. 22 is shown a funnel and spout, which is nothing more than two frustums of cones joined together.

Fig. 23 shows how the patterns are developed. In this figure the full elevation is drawn, but in practice it is necessary to draw only one-half of the elevation, as shown on either side of the center line BC. Extend the contour lines until they intersect the center line at C and A. Now, using A¹ as a center, with radii equal to AF and AE, describe the arcs F¹F² and E¹E² respectively. On the arc E¹E² lay off twice the number of spaces contained in the semi-circle B, then draw radial lines from E¹ and E² to A¹, intersecting the inner arc at F¹F², which completes the outline for the

Fig. 22.

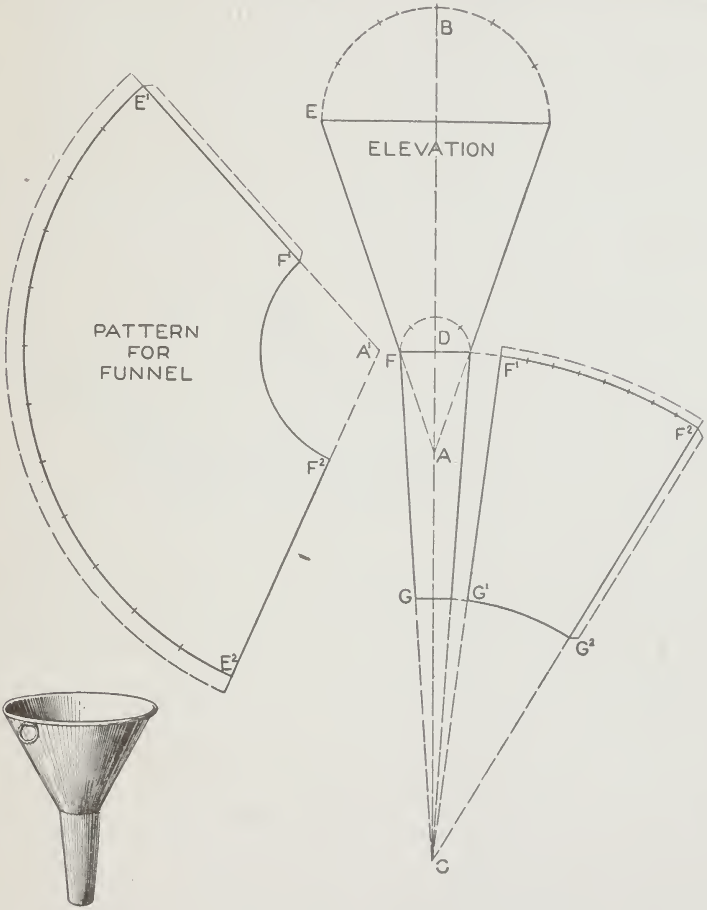


Fig. 23.

pattern. Laps must be allowed for wiring and seaming. For the pattern for the spout use C as a center, and with radii equal to CG and CF describe the arcs F^1F^2 and G^1G^2. On F^1F^2 lay off twice the amount of spaces contained in the semi-circle D, and draw radial lines from F^1 and F^2 to C. Then $F^1F^2G^1G^2$ will be the pattern for the spout. The dotted lines show the edges allowed.

Hand Scoop. In Fig. 24 is shown a perspective view of a hand scoop, in the development of which the parallel and radial line developments are employed. Thus A and B represent intersected cylinders, while C represents an intersected right cone. The method of obtaining the patterns for the hand scoop is clearly shown in Fig. 25; these principles are applicable to any form of hand scoop.

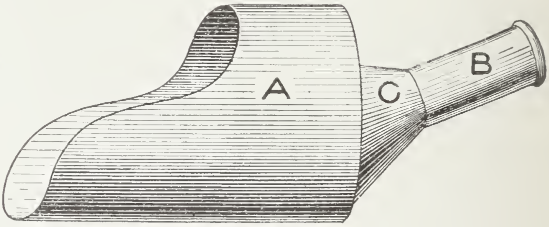


Fig. 24.

First draw the side view of the scoop as shown, in line with which place the half section; divide this into a number of equal spaces as shown by the figures 1 to 7. From these points draw horizontal lines intersecting the curve of the scoop. In line with the back of the scoop draw the vertical line 1–1′, upon which place the stretchout of twice the number of spaces contained in the half section, as shown by similar numbers on the stretchout line. From these points on the stretchout line draw horizontal lines, which intersect lines drawn from similarly numbered points on the curve of the scoop parallel to the stretchout line. Trace a line through points thus obtained, which will give the outline for the pattern for the scoop, to which edges must be allowed as shown by the dotted line. The pattern for the back of the scoop is simply a flat disc of the required diameter, to which edges for seaming are allowed.

When drawing the handle, first locate the point at which the center line of the handle is to intersect the back of the scoop, as at 2°. Through this point, at its proper or required angle, draw the center line $2^{\circ}2^{x}$. Establish the length of the handle, and with any point on the center line as center, draw the section

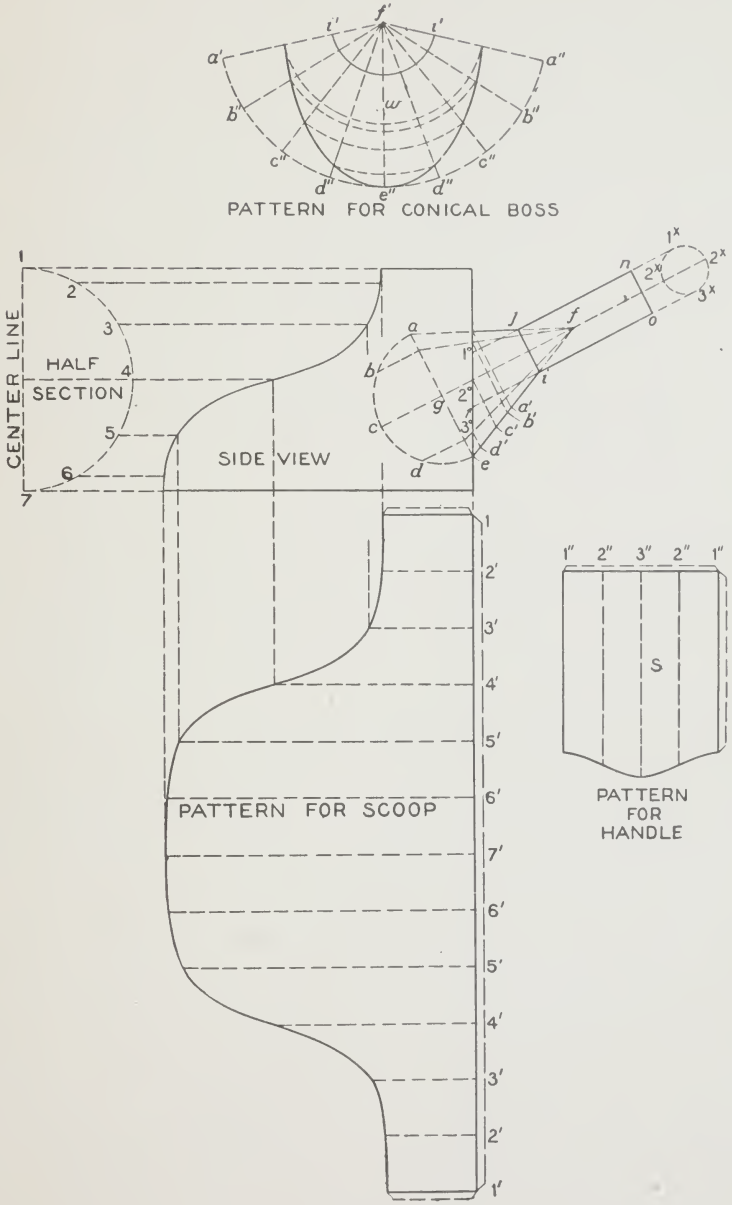

f'
i'
i'
a'
a''
b''
b''
w
c''
c''
d''
e''
d''
PATTERN FOR CONICAL BOSS
1x
2x
2x
3x
n
j
f
o
i
a
b
1°
2°
3°
g
c
d
e
a'
b'
c'
d'
1
2
3
4
5
6
7
CENTER LINE
HALF SECTION
SIDE VIEW
1
2'
3'
4'
5'
6'
7'
6'
5'
4'
3'
2'
1'
PATTERN FOR SCOOP
1'' 2'' 3'' 2'' 1''
S
PATTERN FOR HANDLE

Fig. 25.

as shown by 1^x, 2^x 3^x, and 2^x, and divide the circumference into equal spaces, in this case four. (In practical work it would be better to use more than four). Parallel to the center line and from these four divisions draw lines as shown intersecting the back of the scoop at 1°, 2° and 3°. For the pattern draw any horizontal line in S, as $1''3''1''$, upon which place the stretchout of the section of the handle as shown by $1''$ $2''$ $3''$ $2''$ $1''$ on the stretchout line. From these points at right angles to the line of the stretchout, draw lines as shown. Take the various distances measuring from the line *no* in side view to points 1°, 2° and 3°, and place them on lines drawn from similar numbers in S, measuring from the line $1''3''1''$. A line traced through these points of intersection will be the pattern for the handle, laps being indicated by dotted lines. To close the top of the handle *no*, a small raised metal button is usually employed, which is double-seamed to the handle.

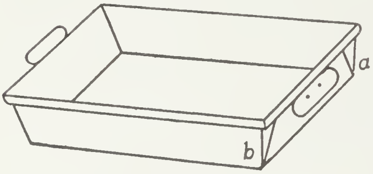


Fig. 26.

To draw the conical boss in side view, first locate the points *i* and *e*, through which draw a line intersecting the center line of the handle at *f*. At right angles to the center line, draw the line *ij* representing the top opening of the boss. In similar manner, at right angles to the center line, draw a line from *e* as shown by *ea*, intersecting the center line at *g*. Now make *ga* equal to *ge* and draw a line from *a* to the center *f*, which will intersect the back of the scoop as shown and the top of the boss at *j*. With *g* as center and *ga* as radius describe the half section of the cone, divide this into equal spaces as shown by *abcde*, from which draw lines at right angles to and intersecting the base of the cone *ae* as shown. From the intersections on the base line draw radial lines to the apex *f*, intersecting the back of the scoop as shown. From these intersections at right angles to the center line, draw lines intersecting the side of the boss at $a'b'c'd'$. For the pattern proceed as shown in diagram *w*. With radius equal to *fe* in the side view and f' in *w*, as a center describe the arc $a'a''$. Draw a line from a'' to the center f', and starting from a'' set off on the arc $a'a''$ twice the number of spaces contained in the semi circle *ace* in side view, as shown by similar letters in diagram *w*. From these points draw radial lines to the center f^3. Now using f^3 in *w* as a center describe the arc $i''i'$. In similar manner, using as radii *fa'*, *fb'*, *fc'*, *fd'* and *fe* in side view, and f^3 in *w* as center, describe arcs intersecting radial lines having similar letters as shown. A line traced through points thus obtained forms the pattern for the conical boss.

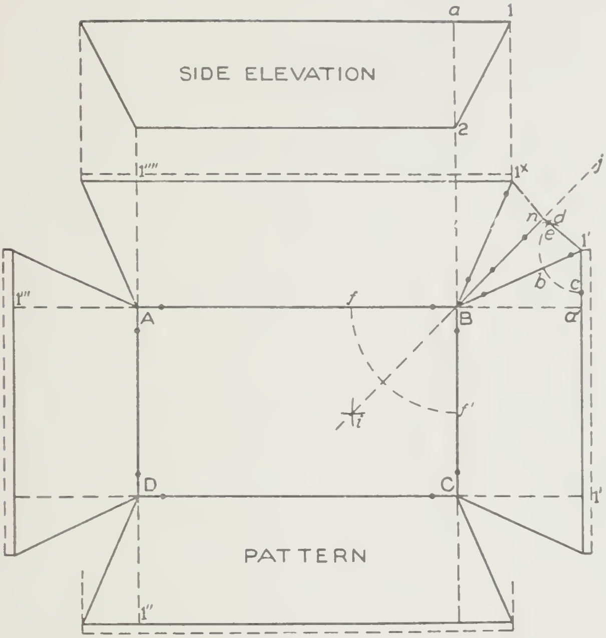


Fig. 27.

Drip Pan. Fig. 26 shows a view of a drip pan with beveled sides. The special feature of this pan is that the corners *a* and *b* are folded to give the required bevel and at the same time have the folded metal come directly under the wired edge of the pan. A pan folded in this way gives a water tight joint without any soldering. Fig. 27 shows the method of obtaining the pattern when the four sides of the pan have the same bevel. First draw the side elevation having a bevel indicated at *a*21. Now draw ABCD, a rectangle representing the bottom of the pan. Take the distance of the slant 1 – 2 in elevation and add it to each side of the rectangular bottom as shown by 1′, 1″, 1‴ and 1⁗. Through these points draw lines parallel to the sides of the bottom as shown. Now extend the lines of the bottom AB, BC, CD and DA intersecting the lines just drawn. Take the projection of the bevel *a* to 1 in side elevation and place it on each corner of the pan, as, for example, from *a′* to 1′. Draw a line from 1′ to B. By proceeding in this manner for all the corners, we will have the butt miters, if the corners were to be soldered raw edge. Where the bevels are equal on all four sides, the angle $1^xB1'$ is bisected as

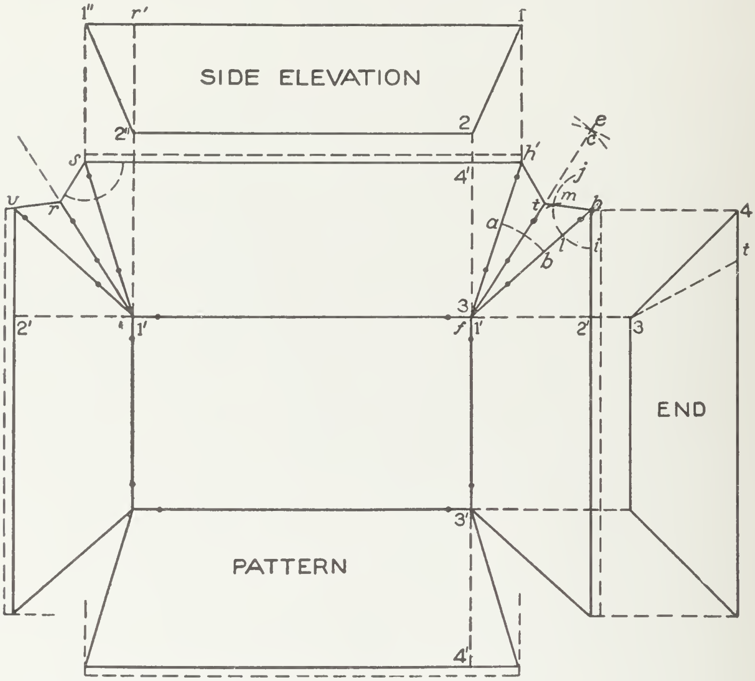


Fig. 28.

follows: With B as center and any radius draw the arc *ff′* intersecting the sides of the bottom as shown. Then with a radius greater than one half of *ff′*, with *f* and *f′* respectively as centers, draw arcs which intersect each other at *i*. Draw a line through the intersection *i* and corner B, extending it outward toward *j*.

Now with 1′ as center, and radius less than one-half of $1'-1^x$, draw arc *d–c*, intersecting the line 1′ B at *b*, and intersecting the line 1′*a′* at *c*. Then with *b* as center and *bc* as radius, intersect the arc *cd* at *e*. Draw a line from 1′ to *e*, intersecting the line *ij* at *n*. From *n* draw a line to 1^x. Transfer this cut to each of the corners, which will complete the pattern desired. Dotted lines indicate the wire allowance.

Sometimes a drip pan is required whose ends have a different flare from those of the sides, and in one case the folded corners are to be bent toward the end, while it may be required that the corners be folded towards the side. The principles are similar in both cases, but as the method of applying these principles may be a little difficult, Fig. 28 has been prepared, which will explain the application of these principles.

First draw the side elevation, showing the desired flare; also draw the end elevation, which shows the flare of the sides, being careful that the vertical heights in both views are the same. Now draw the pattern of the pan as follows: Take the distance 1–2 in side elevation and place it on the ends of the bottom as shown on either side by 1′–2′. Similarly take the distance 3–4 in end elevation and place it on the sides of the bottom as shown on either side by 3′–4′. Through the point 2′ and 4′ draw lines parallel to the ends and sides of the bottom as shown, which intersect lines dropped from the end and side views respectively. hfh' represent the butt miters which should be placed on all corners. If these miters have been correctly developed, the lengths from h to f must be equal to fh'. Bisect the angle hfh' by using f as center and drawing the arc ab, then use a and b as centers and obtain the intersection c, through which draw the line cf. Now assume that the folded corner is to be turned towards the end view as shown by $t3$. Using h as a center draw the arc ij. Then with l as center and li as radius, intersect the arc ij at m. Draw a line from h through m, meeting the line cf at t, and draw a line from t to h'.

If the folded corner were turned towards the side as shown by r'–2″ in the side view, bisect the angle $v1's$ as before, and use s as a center and proceed as already explained. Note the difference in the two corners. The only point to bear in mind is, that when the corner is to be folded towards the end, transfer the angle of the end miter; while if the corner is to be turned towards the side, transfer the angle of the side miter. If the corners were to be folded toward the ends of the pan, the cut shown in the right-hand corner would be used on all four corners, while if the corners were to be folded towards the sides, the cut shown on the left-hand corner would be used.

Tea Pot. In Fig. 29 is shown the well-known form of the tea or coffee pot, for which a short method of developing the pattern is shown in Fig. 30. This is one of the many cases where a short rule can be used to advantage over the geometrical method. While it is often advisable to use the true geometrical rule, the difference between that and the method here shown is hardly noticeable in practice. Of course, if the body A and spout B were larger than the ordinary tea pots in use, it would be necessary to use the true geometrical rule, which is thoroughly explained for Plates I, II and III.

Fig. 29.

The pattern for the body of the tea pot will not be shown, only the short rule for obtaining the opening in the body to admit the joining of the spout. The method of obtaining the pattern for the body is similar to the flaring vessels shown in previous problems.

First draw the elevation of the body of the tea pot as shown at A. Assume the point *a* on the body and draw the center line of the spout at its proper angle as shown by 2*b*. Establish the point 3 of the bottom of the spout against the body, also the point 3^x at the top and draw a line from 3 through 3^x intersecting the center line at *b*. At right angles to the center line and from 3 draw the line 3 – 1 and make *c*1 equal to *c*3. From 1 draw a line to the center point and from 3^x draw a horizontal line until it intersects the opposite side of the spout at 1″. Then 1′ – 1″ – 3^x–3 will be the side view of the spout. Now with *c* as a center draw the half section 1–2–3 and divide it into equal spaces; in this case but two (in practical work more spaces should be employed). From these points and at right angles to 1 – 3 draw lines intersecting the base of the spout as shown, and draw lines from these points to the center *b*. Thus line 1*b* intersects the body at 1′ and the top of the spout at 1″; line 2*b* intersects the body at *a* and the top of the spout as shown, while line 3*b* cuts at 3 and the top of spout at 3^x. From these intersections at right angles to the center line *ab*, draw lines intersecting the side of the spout at 3, 2°, 1° at the bottom and 1^x, 2^x, 3^x at the top. Now with *b* as center and *b*3 as radius, describe the arc 3′ – 3″ upon which place the stretchout of twice the number of spaces contained in the half section 1 – 2 – 3, as shown by similar figures on 3″ – 3″; from these points draw radial lines to the center *b*, and intersect them by arcs drawn with *b* as a center and radii equal to the intersections contained on the side of

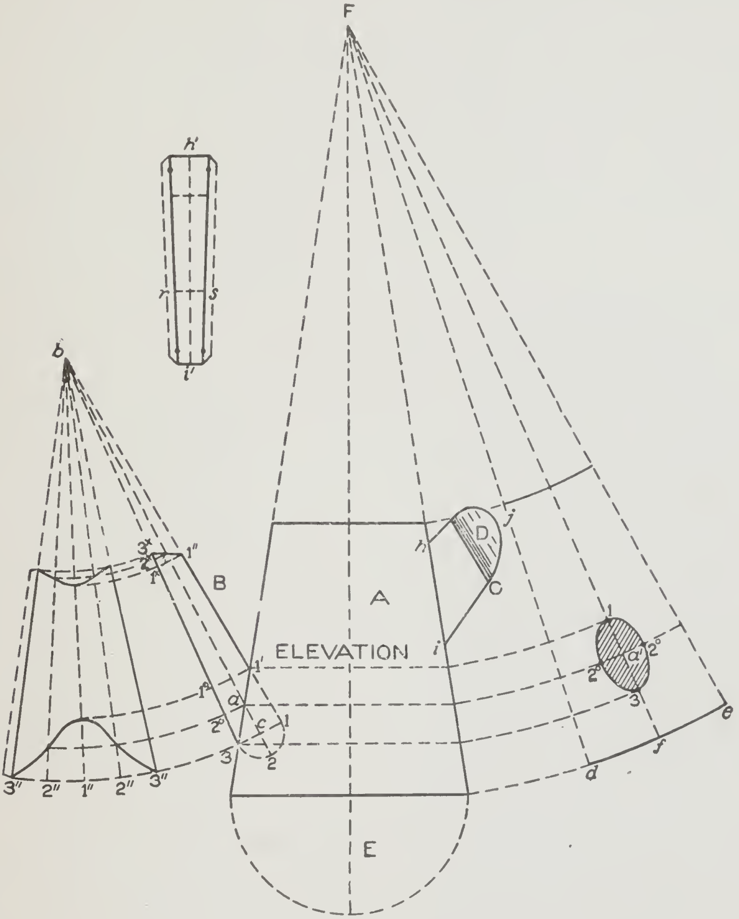


Fig. 30.

the spout 3 – 3ˣ. To form the pattern, trace a line through points thus obtained and make the necessary allowance for edges.

It should be understood that in thus developing the spout, the fact that the spout intersects a round surface has not been considered; it was assumed to intersect a plane surface. As already stated the difference in the pattern is so slight that it will not be noticeable in practice. Had we developed the pattern according to the true geometrical rule, it would present a problem of two cones of unequal diameter intersecting each other, at other than at right angles to the axes.

For the pattern for the opening in the body, draw lines at right angles to the center line of the body from intersections 1′, *a* and 3 intersecting the opposite side of the body as shown. With F as a center draw a partial pattern of the body as shown by *de*. From any point *f* draw a line to the center F. Now with F as center draw the arcs 1, 2° and 3. The distance 1 to 3 on the line F*f* represents the length of the opening, while a line drawn through *a* at right angles to the center line *bc* of the spout represents the width of the opening. Therefore take the distance from *a* to 2° and place it as shown from *a′* on the line *f*F to 2° – 2° on either side on the arc. Trace an ellipse through 1 – 2° – 3 – 2° for the shape of the opening.

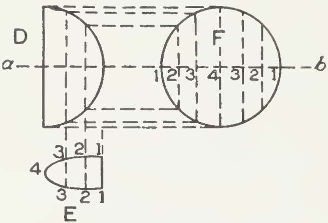


Fig. 31.

The pattern for the handle is obtained by taking the stretchout of *hji* and placing it as shown on the vertical line *h′i′*. At right angles to *h′i′* on either side, at top and bottom add the desired width of the handle and draw the lines shown; add edges for wiring or hem edge.

For the pattern for the grasp D which is placed inside on the handle proceed as is shown in Fig. 31. Let D represent an enlarged view of part of the handle in which the grasp is to be soldered. Directly in line with it draw the section E taking care that the width from 1 to 1 will not be wider than that portion of the handle from *r* to *s* in Fig. 30, being the width at C in the elevation. Divide the section E in Fig. 31 into a number of equal spaces, from which draw vertical lines intersecting the curve D as shown. Draw the center line *ab* upon which lay off the stretchout of E as shown by similar figures. Through these points draw lines which intersect with lines drawn from similar intersections in the curve D parallel to *ab*. Trace a line through the points thus obtained as shown at F.

Foot Bath. In Fig. 32 is shown an oval foot bath; the principles used in obtaining the pattern of which are applicable to any form of flaring vessels of which the section is elliptical or struck from more than two centers. In this connection it may be well to explain how to construct an ellipse, so that a set of centers can be obtained with which to strike the arcs desired. Fig. 33 shows the method of drawing an approximate ellipse, if the dimensions are given. Let AB represent the length of the foot bath and CD its width. On BA measure BE equal to CD. Now divide the distance EA into three equal parts as shown by 1 and 2. Take two of these parts as a radius, or E2, and with O as center, describe arcs intersecting the line BA at X and X^1. Then with XX^1 as a radius and using X and X^1 as centers describe arcs intersecting each other at C and D. Draw lines from C to X and X^1 and extend them toward F and G respectively. Similarly from D draw lines through X and X^1, extending them towards I and H respectively. Now with X and X^1 as centers, and XA and X^1B as radii describe arcs intersecting the lines ID, FC, GC and HD at J, K, L and M, respectively. In similar manner with D and C as centers and DC and CD as radii describe arcs which must meet the arcs already drawn at J, M, L and K, respectively, forming an approximate ellipse. In Fig. 34 let ABCD represent the side elevation of the pan, whose vertical height is equal to RC.

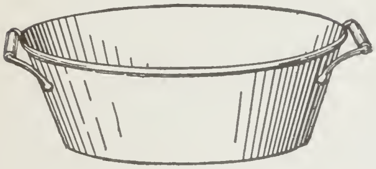


Fig. 32.

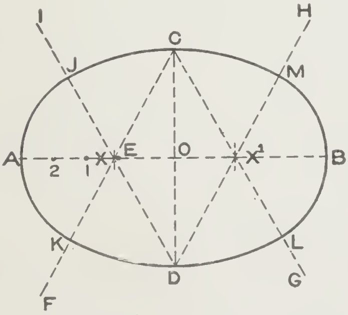


Fig. 33.

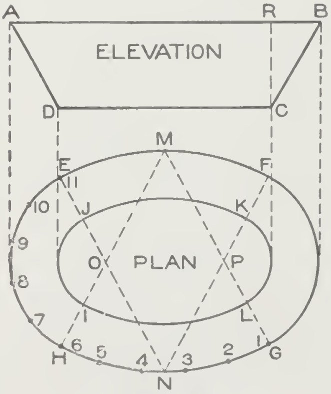


Fig. 34.

In precisely the same manner as described in Fig. 33 draw the plan as shown, in correct relation to the elevation, letting EFGH be the plan of the top of the pan, and JKLI the plan of the bottom, struck from the centers, O,M,P and N. The next step is to obtain the radii with which to strike the pattern. Draw a horizontal line RE in Fig. 35 equal in length to NE in plan in Fig 34. Take the vertical height RC in elevation, and place it as shown by RC in Fig. 35 on a line drawn at right angles to RE. Parallel to RE and from the point C, draw the line CJ equal to NJ in Fig. 34.

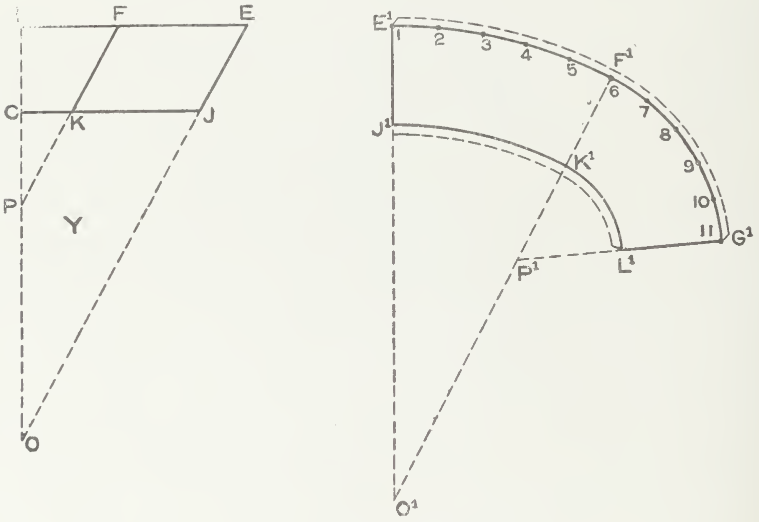


Fig. 35.

Now draw a line from E to J in Fig. 35, extending it until it meets the line RC produced. Then OJ and OE will be the radii with which to make the pattern for that part of the pan or foot bath shown in plan in Fig. 34 by EFKJ and GHIL.

To obtain the radii with which to strike the smaller curves in plan, place distances PF and PK on the lines RE and CJ in Fig. 35 as shown by RF and CK. Draw a line from F through K until it meets the line RO at P. Then PK and PF will be the radii with which to strike the pattern, for that part shown in plan in Fig. 34 by KFGL and IHEJ. Now divide the curve from G to H and H to E (Fig. 34) into a number of equal spaces. To describe the pattern draw any vertical line E^1O^1 (Fig. 35) and with O^1 as center and radii equal to OJ and OE in the diagram Y, describe the arcs J^1K^1 and E^1F^1 as shown. On the arc E^1F^1 lay off the stretchout of GH in plan in Fig. 34 as shown by similar figures in Fig. 35. From the point 6 on the arc E^1F^1 draw a line to O^1 intersecting the curve J^1K^1. Now with PF in diagram Y as radius and F^1 as a center describe an arc intersecting the line F^1O^1 at P^1. Then using P^1 as a center and with radii equal to P^1K^1 and P^1F^1 describe the arcs K^1L^1 and F^1G^1 as shown. On the arc F^1G^1 starting from point 6 lay off the stretchout of HE. Fig. 34. From 11 draw a line to P^1 intersecting the arc K^1L^1 at L^1. Then $E^1F^1G^1L^1K^1J^1$ will be the half pattern, the allowance for wiring and seaming being shown by the dotted lines.

Should the article be desired in four sections, two pieces of $F^1K^1L^1G^1$ would be required. The pattern for the bottom of the pan is shown by the inner ellipse in Fig. 34 to which of course edges must be allowed for double seaming.

Fig. 36.

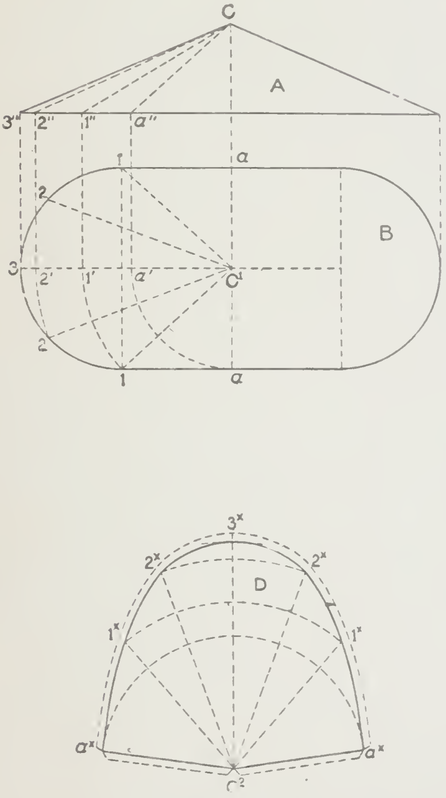


Fig. 37.

Wash Boiler. In Fig. 36 is shown a perspective view of a wash boiler to which little attention need be given, except to the raised cover. First draw the plan of the cover B, Fig. 37, which shows straight sides with semi-circular ends. In line with the plan draw the elevation A, giving the required rise as at C. Let C represent the apex in elevation, and C^1 the apex in plan. As both halves of the cover are symmetrical, the pattern will be developed for one half only. Divide the semi-circle 1 – 3 – 1 into a number of equal spaces as shown by the small figures 1, 2, 3, 2 and 1. From these points draw radial lines to the apex C^1, and through C^2 draw the perpendicular *aa*. $C3''$ in elevation represents the true length of $C^1 3$ in plan, and to obtain the true length of $C^1 2$, $C^1 1$ and $C^1 a$, it will be necessary to construct a diagram of triangles as follows: With C^1 as center, and $C^1 a$, $C^1 1$ and $C^1 3$ as radii, describe arcs intersecting the center line in plan at a', $1'$ and $2'$. From these points at right angle to $3C^1$ erect lines intersecting the base line of the elevation at a'', $1''$, $2''$ and $3''$, from which draw lines to the apex C, as shown. Now, with radii equal to $C3''$, $C2''$, $C1''$ and Ca'', and C^2 as center describe arcs 3^x,2^x2^x, 1^x1^x and a^xa^x. From C^2 erect the perpendicular intersecting the arc 3^x at 3^x. Now set the dividers equal to the spaces 3 to 2 to 1 to *a* in plan, and starting from 3^x step off to similar numbered arcs, thus obtaining the intersections $2^x1^xa^x$; from a^x draw lines to C^2, and trace a line $a^x3^xa^x$ to get the half pattern for the cover. Allow edges for seaming.

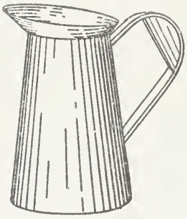


Fig. 38.

The body of the boiler requires no pattern, as that is simply the required height, by the stretchout of the outline shown in plan. The handles shown on the body and cover in Fig. 36 are plain strips of metal to which wired or hem edges have been allowed.

Measure. Fig. 38 shows a flaring-lipped measure with handle attached. Care should be taken in laying out the patterns for these measures, that when the measure is made up it will hold a given quantity. While there are various proportions used in making up the size of the measure, the following table gives good proportions:

Quantity.	Bottom Diameter in inches.	Top Diameter. in inches.	Height.
1 Gill.	2.06	1.37	3.10
½ Pint.	2.60	1.75	3.89
1 Pint.	3.27	2.18	4.90
1 Quart.	4.12	2.75	6.18
½Gallon.	5.18	3.45	7.78
1 Gallon.	6.55	4.35	9.80

Fig. 39 shows the method of laying out the pattern for the measure and lip. First draw the elevation A to the desired size and assume the flare of the lip B, as shown by *db*. From *b* draw a line through 7″ to *c* which is a chosen point, and draw a line from *c* to *d*. Draw the handle C of the desired shape. Now extend contour lines of the measure until they intersect at *a*, and draw

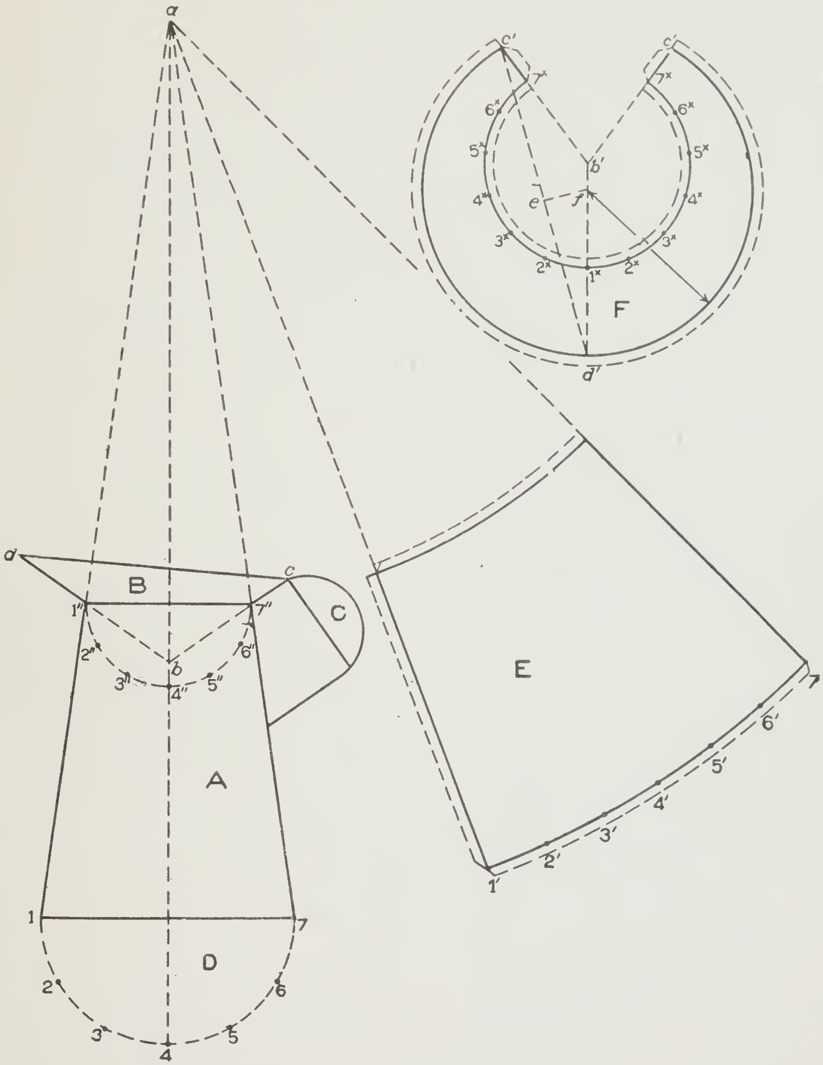


Fig. 39.

the half section of the bottom of the measure as shown at D; divide this semi-circle into equal parts as shown. Now, with a as a center, and $a7$ and $a7''$ as radii, describe the arcs as shown. From any point (as $1'$) draw a radial line to a, and starting at $1'$ set off the number of spaces contained in the half section D, as shown by the small figures $1'$ to $7'$. From $7'$ draw a radial line to a. Allow edges for wiring and seaming. E represents the half pattern for the body of the measure. We find that lip B is simply an intersected frustum of a right cone, which can be developed as shown in the pattern for conical boss of Fig. 25.

There is, however, a shorter method which serves the purpose just as well; this is shown at F, Fig. 39. First draw the half section of the bottom of the lip, which will also be the half section of the top of the measure, as shown by the figures $1''$ to $7''$. Now, with radii equal to $b-1''$, or $b-7''$ and b' in F as center, describe the arc 7^x7^x. From b' drop a vertical line intersecting the arc at 1^x. Starting from the point 1^x, set off the spaces contained in the half section $1''-4''-7''$, as shown by the figures 1^x to 7^x. From b' draw lines through the intersections 7^x7^x, extending them as shown. Now take the distance from $1''$ to d of the front of the lip and place it as shown by $1^xd'$ in F. In similar manner take the distance from $7''$ to c of the back of the lip and place it as shown in F from 7^x to c' on both sides. Draw a line from c' to d', and bisect it to obtain the center e. From e, at right angles to $c'd'$, draw a line intersecting the line $b'd'$ at f. Then using f as center, with radius equal to fd', draw the arc $c'd'c'$, as shown. Adding laps for seaming and wiring will complete the pattern for the lips.

The pattern for the handle and grasp C is obtained as shown in Figs. 30 and 31.

Scale Scoop. Fig. 40 shows a scale scoop, wired along the top edges and soldered or seamed in the center. The pattern is made as shown in Fig. 41. First draw the elevation of the scoop as shown by ABCD. (In practice the half elevation, BDC, is all that is necessary.) At right angles to BD and from the point C, draw the indefinite straight line CE, on which a true section is to be drawn. Therefore, at right angles to CE, from points C and E, draw the lines CC and EE^1. From E^1 erect a perpendicular as E^1C, on which at a convenient point locate the center F; with

FE^1 as radius, describe the arc HE^1I. Then HE^1I will be the true section on CE in elevation. Divide the section into a number of equal parts as shown by the figures 1 to 7; through these points, parallel to the line of the scale BD, draw lines intersecting BC and CD as shown. At right angles to BD draw the stretchout line 1 – 7 and place upon it the stretchout of the section as shown by similar figures. At right angles to 1 – 7 draw lines which intersect lines drawn at right angles to BD, from intersections on BC and DC having similar numbers. Trace a line through these points and thus obtain the desired pattern. The dotted outline shows the lap and wire allowance.

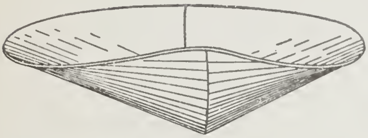


Fig. 40.

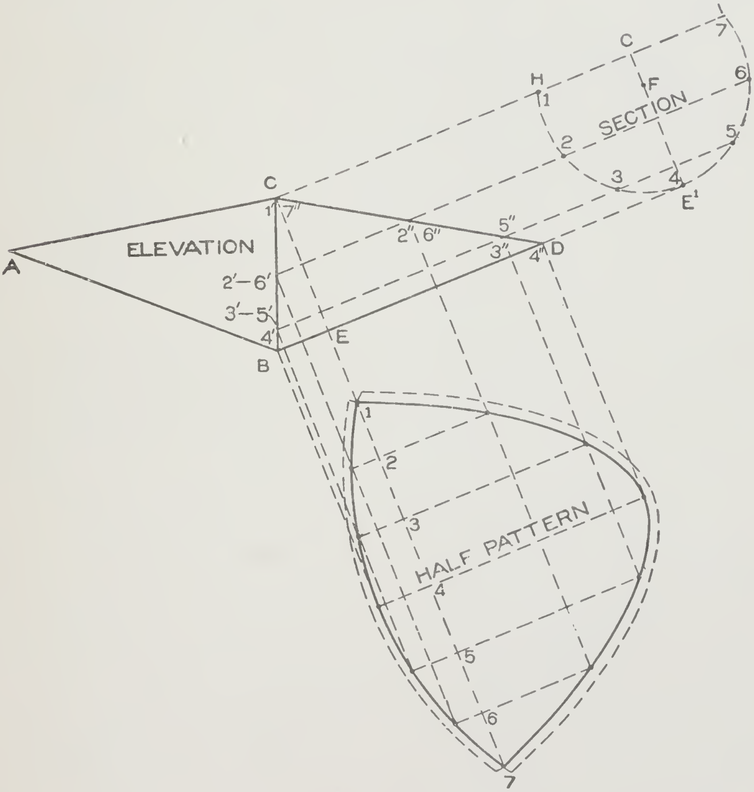


Fig. 41.

In Fig. 42 is shown a perspective view of a **dust pan** with a tapering handle passing through the back of the pan and soldered to the bottom. The first step is to draw the plan and elevation which is shown in Fig. 43. Let ABC be the side view of the pan. Directly below it, in its proper position, draw the bottom DEFG. From the point C in elevation draw a line $d'd$ indefinitely. Now bisect the angle EFG. Through e and F draw the line ed, intersecting the line dd' at d. From d draw a line to G.

In the same manner obtain Ed'D on the opposite side, which

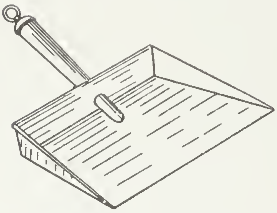


Fig. 42

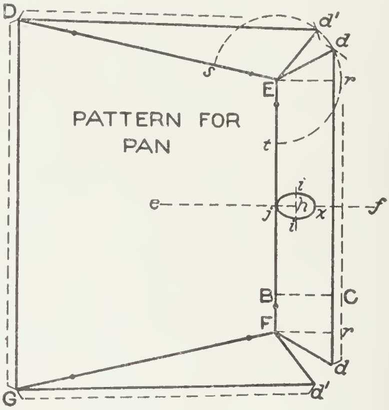


Fig. 44.

will complete the plan view of the pan. Now locate the point h in side view, through which the center line of the handle shall pass, and draw the line lm Through m, the end of the handle, draw the line no at right angles to lm, and assume o the half width at the top and j the point where the contour line of the handle shall meet the back of the pan, and draw a line from o through j, intersecting the center line lm at l. Make mn equal to mo and draw a line from n to l, intersecting the back of the pan at x. Through h at right angles to the center line draw ij'', giving the diameter of the handle at that point to be used later. This completes the elevation of the handle; the plan view is shown by dotted lines and similar letters, but is not required in developing the pattern.

For the pattern of the pan proceed as is shown in Fig. 44, in which DEFG is a reproduction of similar letters of Fig. 43. Take the distance of BC in side view, Fig. 43, and place it as shown by

BC in Fig. 44 and through C draw a line parallel to EF as shown. At right angles to and from EF draw E*r* and F*r*, then take the distance from *r* to *d* in plan in Fig. 43 and place it as shown from *r* to *d* on both sides in Fig. 44. Draw the lines *d*F and *d*E. Now using E as center, and radius equal to E*d* describe the arc *st*. Then with *td* as radius and *s* as center, intersect the arc *st* at *d'*. Draw a line from *d'* to D. In similar manner obtain *d'*G on the opposite side, which will complete the pattern for the pan. Allow laps for wiring and edging.

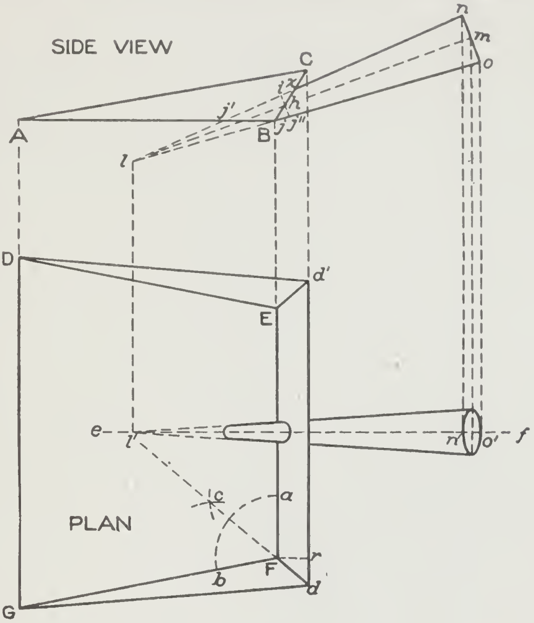


Fig. 43.

The opening in the back of the pan to allow the handle to pass through is obtained by first drawing a center line *ef*, then take the distances from *j* to *h* and *h* to *x* in Fig 43, noting that *j* comes directly on the bend B, and place it in Fig. 44 on the line *ef*

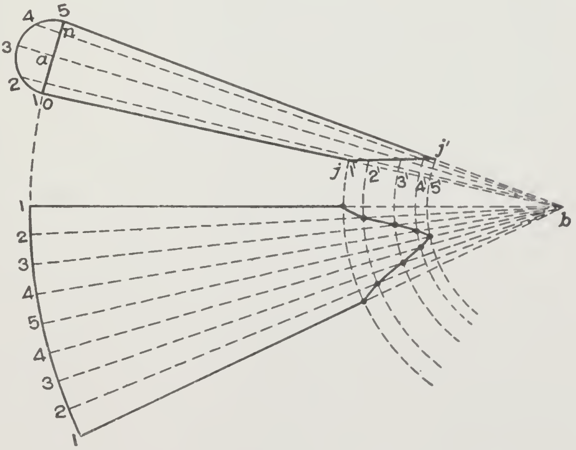


Fig. 45.

from j to h to x, placing j on the bend as shown. Now take the distance from h to i or h to j'' in side view in Fig. 43 and place it in Fig. 44 from h to i on either side; on a line drawn through the points $jixi$ draw an ellipse shown. Fig. 45 shows the method of drawing the pattern for the tapering handle. From the figure we find that we have a frustum of a right cone. To illustrate each step the handle has been slightly enlarged. n, o, j, j' represents n, o, j, j' in Fig. 43. Draw the half section in Fig. 45 as shown, and divide it into equal parts; drop perpendiculars as shown to the line no, and from these points draw lines to the apex b, which is obtained by extending the lines nj' and oj until they meet at b. Where the radial lines intersect the line jj' draw lines at right angles to the center line 3b, intersecting the side of the handle o b at 1′, 2′, 3′, 4′ and 5′. Now with b as a center and bo as a radius, describe the arc 1–1, upon which place twice the number of spaces contained in the half section a. From these points on 1–1 draw radial lines to b and cut them with arcs struck from b as center and radii equal to $b1'$, $b2'$, $b3'$, $b4'$ and $b5'$. Trace a line through points thus obtained to complete the pattern.

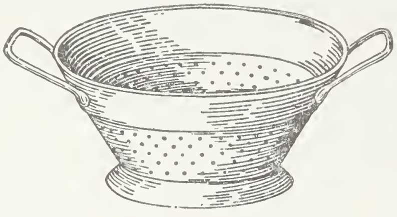


Fig. 46.

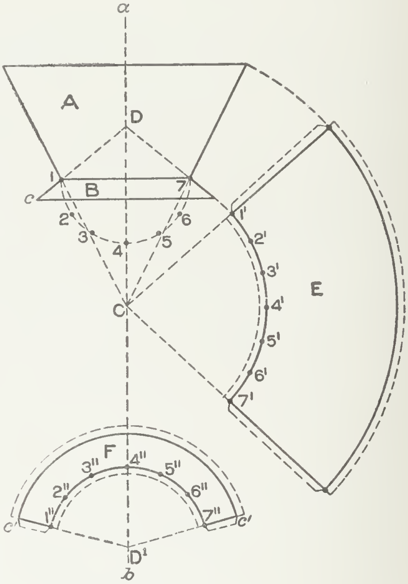


Fig. 47.

Colander. Fig. 46 shows another well-known form of tin ware, known as a colander. The top and bottom are wired and the foot and body seamed together, the handles of tinned malleable iron being riveted to the body. In Fig. 47 is shown how to lay out the patterns. Draw the elevation of the body A and foot B and extend the sides of the body and foot until they meet respectively at C and D on the center line *ab*. Draw the half section on the line 1–7 and divide it into equal parts as shown. For the body, use C as a center and describe the arcs shown, laying off the stretchout on the lower arc, allowing edges in the usual manner. Then E will be the half pattern for the body. In the usual manner obtain the pattern for the foot shown at F, the pattern being struck from D^1 as center, with radii obtained from the elevation D1 and D*c*.

TABLE OF STANDARD OR REGULAR TIN PLATES.

Size and Kind of Plates, Number and Weight of Sheets in a Box, and Wire Guage Thickness, of Every Kind and Size.

Size.	Grade.	Sheets in Box.	Pounds in Box.	Wire Guage.
10 x 10	IC	225	80	29
"	IX	225	100	27
"	IXX	225	115	26
"	IXXX	225	130	25
"	IXXXX	225	145	24 1-2
10 x 14	IC	225	112	29
"	IX	225	140	27
"	IXX	225	161	26
"	IXXX	225	182	25
"	IXXXX	225	203	24 1-2
"	IXXXXX	225	224	24
"	IXXXXXX	225	245	23 1-2
10 x 20	IC	225	160	29
"	IX	225	200	27
11 x 11	IC	225	95	29
"	IX	225	121	27
"	IXX	225	139	26
"	IXXX	225	157	25
"	IXXXX	225	175	24 1-2
11 x 15	SDC	200	168	26
"	SDX	200	189	25
"	SDXX	200	210	24 1 2
11 x 15	SDXXX	200	230	24
12 x 12	IC	225	112	29
"	IX	225	140	27
"	IXX	225	161	20
"	IXXX	225	182	25
"	IXXXX	225	203	24 1-2
"	IXXXXX	225	224	24
"	IXXXXXX	225	245	23 1-2
12 1-2 x 17	DC	100	98	28
"	DX	100	126	26
"	DXX	100	147	24
"	DXXX	100	168	23
"	DXXXX	100	189	22
"	DXXXXX	100	210	21
13 x 13	IC	225	135	29
"	IX	225	169	27
"	IXX	225	194	26
"	IXXX	225	220	25
"	IXXXX	225	245	24 1-2
13 x 17	IXX	225	254	26
13 x 18	IX	225	234	27
"	IXX	225	269	26
14 x 14	IC	225	157	29
"	IX	225	196	27
"	IXX	225	225	26
"	IXXX	225	255	25

TABLE OF STANDARD OR REGULAR TIN PLATES.===Con.

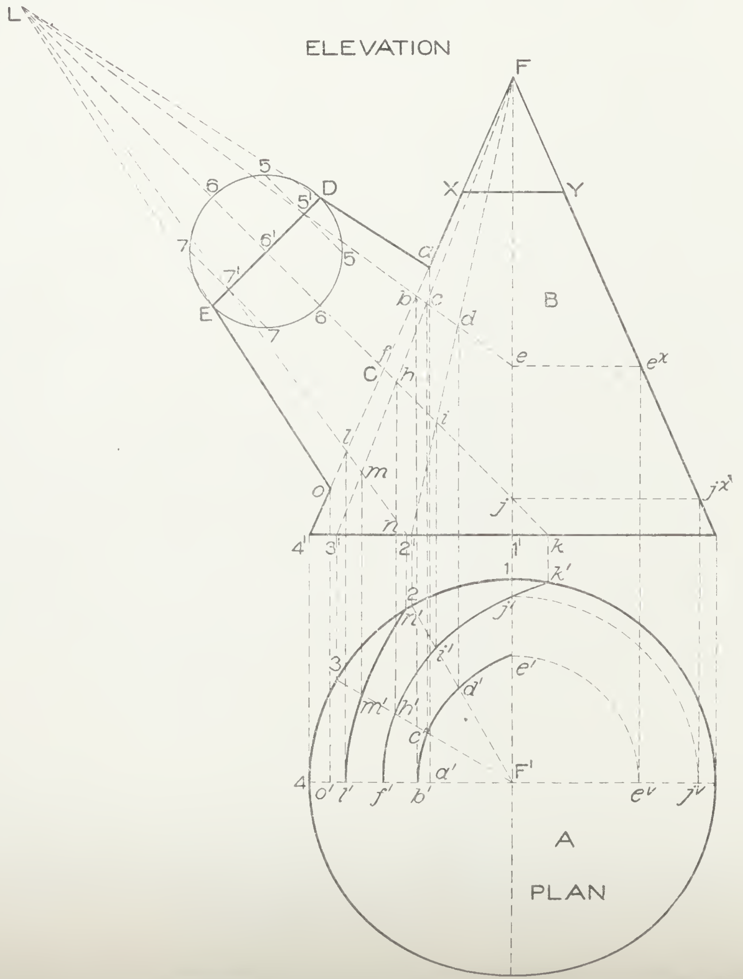
Size.	Grade.	Sheets in Box.	Pounds in Box.	Wire Gauge.
14 x 14	IXXXX	255	284	24 1-2
14 x 17	IX	225	238	27
14 x 20	IC	112	113	29
“	IX	112	143	27
“	IXX	112	162	26
“	IXXX	112	183	25
“	IXXXX	112	202	24 1-2
15 x 15	IX	225	225	27
“	IXX	225	259	26
“	IXXX	225	293	25
“	IXXXX	225	326	24 1-2
15 x 21	IX	112	158	27
“	DXX	100	218	24
“	DXXX	100	249	23
“	DXXXX	100	280	22
15 x 22	IXX	112	190	26
“	SDXX	100	210	24 1-2
“	SDXXX	100	230	24
16 x 16	IC	225	205	29
“	IX	225	256	27
“	IXX	225	294	26
“	IXXX	225	333	25
“	IXXXX	225	371	24 1-2
17 x 17	IC	225	231	29
17 x 17	IX	225	289	27
“	IXX	112	166	26
“	IXXX	112	188	25
“	IXXXX	112	210	24 1-2
17 x 25	DC	100	196	28
“	DX	100	252	26
“	DXX	50	146	24
“	DXXX	50	168	23
“	DXXXX	50	189	22
“	IX	112	213	27
“	IXX	112	244	26
18 x 18	IX	112	162	27
“	IXX	112	186	26
“	IXXX	112	211	25
“	IXXXX	112	235	24 1-2
19 x 19	IC	112	144	29
“	IX	112	180	27
“	IXX	112	207	26
“	IXXX	112	234	25
“	IXXXX	112	262	24 1-2
20 x 20	IC	112	160	29
“	IX	112	200	27
“	IXX	112	230	26
“	IXXX	112	260	25
“	IXXXX	112	290	24 1-2
20 x 28	IC	112	224	29
“	IX	112	280	27
“	IXX	112	322	26

TERNE PLATES.

Size.	Grade.	Sheets in Box.	Pounds in Box.	Wire Gauge.
14 x 20	IC	112	112	29
"	IX	112	140	27
20 x 28	IC	112	224	29
"	IX	112	280	27
20 x 200	IC	Roll	176	29
"	IX	"	220	27

EXAMINATION PAPER

PLATE I


ELEVATION
L
F
X
Y
D
E
5
6
7
5'
6'
7'
a
b
c
d
e
e^x
B
C
f
h
i
l
m
o
n
j
$j^{x'}$
4'
3'
2'
1'
k
1
2
3
4
k'
j'
n'
i'
e'
d'
m'
h'
c'
a'
F'
o'
l'
f'
b'
e^v
j^v
A
PLAN

EXAMINATION PLATES.

Drawing Plates I to IV inclusive constitute the examination for this Instruction Paper. The student should draw these plates in ink and send them to the School for correction and criticism. The construction lines and points should be clearly shown. The date, student's name and address, and plate number should be lettered on each plate in Gothic capitals.

In preparing the plates, the student should practice on other paper, and then send finished drawings for examination. The plates of this instruction paper should be laid out in the same manner and of the same size as the plates in Mechanical Drawing Parts I, II and III.

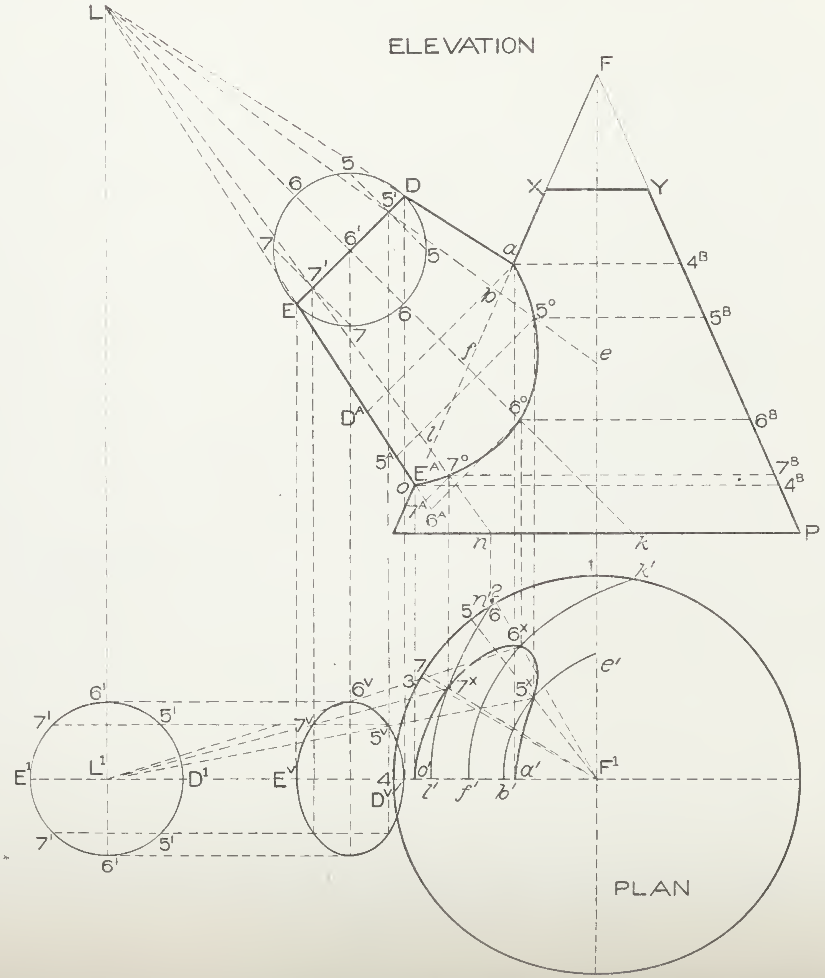
PLATE I.

On this plate, the intersection between two right cones is shown. This problem arises in the manufacture of tinware in such instances as the intersections between the spout and body as in a teapot, watering pot, kerosene-oil can, dipper handle and body, and other articles. It is one of the most complicated problems arising in tinsmith work. The problem should be drawn in the center of the sheet making the diameter of the base A 4 inches and the height of the cone B $4\frac{1}{2}$ inches. The distance from X to Y should be 1 inch. From the point F measure down on the side of the cone a distance of $3\frac{1}{8}$ inches and locate the point C, from which draw the axis of the smaller cone at an angle of 45° to the axis of the larger cone. From C measure on CL $1\frac{5}{8}$ inches locating the point 6′; through this point, at right angles to the axis, draw ED equal to $1\frac{1}{2}$ inches. From the point 4′ on the base of the cone, measure up on the side of the cone a distance of $\frac{1}{2}$ inch as indicated by o, and draw a line from o to E, extending it, until it intersects the axis LC at L. From L draw a line through D extending it until it intersects the larger cone at a. Then D a o E will represent the outline of the frustum of the smaller cone in elevation.

The next step is to obtain the line of intersection between the two cones, but before this can be accomplished, horizontal sections must be made through various planes of the smaller cone cutting into the larger. As the intersection of each half of the smaller cone with the larger one is symetrical, and as the small cone will not intersect the larger one to a depth greater than the point 1 in plan, divide only one-quarter of the plan into a number of equal spaces as shown by figures 1 to 4; from these points draw radial lines to the center F^1 as shown. Also from points 1, 2, 3 and 4 erect vertical lines intersecting the base of the cone at 1′, 2′, 3′ and 4′ respectively, from which points draw radial lines to the apex F.

Now with 6′ on the line ED as a center describe the circle shown, which represents the true section on ED. Divide each semi-circle into the same number of divisions as shown by the small figures D, 5, 6, 7, and E on either side. From these points at right angles to ED draw lines intersecting the center line ED at 5′. 6′ and 7′. From the apex L draw lines through the intersection 5′, 6′ and 7′, and extend them until they intersect the axis of the large cone at *e* and the base line at *k* and *n*. The student may naturally ask why the radial lines in the small cone are drawn to these points. As it is not known how far the smaller cone will intersect the larger one, we obtain such sections on the planes just drawn, as we think will be required to determine the depth of the intersection. Thus the radial line drawn through 5′ intersects the radial lines through 4′, 3′, 2′ and 1′ in the larger cone, at *b*, *c*, *d* and *e* respectively. The radial line through 6′ intersects radial lines in the larger cone at *f*, *h*, *i*, *j* and the base line at *k*, while the radial line drawn through the point 7′, intersects the radial lines of the larger cone at *l* and *m* and the base at *n*. We know that the line D*a* and E*o* of the smaller cone intersect the larger cone at points *a* and *o* respectively, and determine the true points of intersection; these are shown in plan by *a*′ and *o*′, and therefore no horizontal sections are required on these two planes. For the horizontal section on the plane *b e*, drop vertical lines from the intersections *b*, *c* and *d* on the radial lines, intersecting radial lines having similar numbers in plan as shown by *b*′, *c*′ and *d*′. To obtain the point of intersection in plan of *e* in elevation, draw from the point *e* a horizontal line intersecting the side of the cone at e^x, from which point drop a perpendicular line intersecting the center line in plan at e^v.

PLATE II


ELEVATION
L
F
X
Y
D
E
5
6
7
5'
6'
7'
a
b
e
f
l
5°
6°
7°
4B
5B
6B
7B
4B
DA
5A
EA
o
7A
6A
n
k
P
PLAN
1
k'
n
2
5
6
6x
7
3
7x
5x
e'
F1
a'
b'
f'
l'
o'
4
DV
EV
6V
5V
7V
6'
7'
5'
E1
L1
D1
7'
5'
6'

Then using $F^1 c^v$ as radius, describe an arc intersecting the radial line 1 at c'. Through the points c', d', e' and b' draw a curved line, which is the half horizontal section of $b\ c$ in elevation. In the same manner obtain the sections shown in plan by f', h', i', j' and k'; and l', m' and n', which have similar letters and figures in both plan and elevation. The next step is to obtain the intersections where the radial lines of the smaller cone will intersect these sections in plan just obtained. To avoid a confusion of lines which would otherwise occur, a reproduction of the plan and elevation has been transferred to Plate II.

PLATE II.

The figures on this plate are similar to those on Plate I and have similar letters and figures; those letters and figures being omitted which are not necessary. This plate should be studied carefully before proceeding. The reproducing from Plate I can be best done by using a needle point or the small needle which is usually found in the handle of the drawing pen, when unscrewing the pen from the handle, and pricking through Plate I, very small indistinct prick marks. Omit all that is omitted in Plate II, where it will be noticed that the radial line in elevation, of the larger cone, and some of the various small letters in plan are not represented.

To obtain the plan view of the smaller cone, proceed as follows: Extend the line $F^1\ 4$ in plan as shown by $F^1\ E^1$. From the apex L of the smaller cone drop a vertical line intersecting $F^1\ E^1$ at L^1, which represents the apex of the smaller cone in plan. With L^1 as center and radius equal to the radius $6'\ D$ describe the circle $E^1\ D^1$ and divide the circumference into the same number of spaces as ED, being careful to number the spaces as is there shown. The reason for doing this may be better understood from what follows: Assume that ED is a pivot on which the circle turns, so that it lies on a plane ED, then by looking down from the top, the points 6 and 6 appear as shown by $6'$ and $6'$ in plan.

A better illustration is obtained by cutting a card-board disc and after spacing it and numbering the points hold it in various positions until all the points become clear. Now from the intersections on ED in elevation, drop lines, intersecting horizontal lines drawn from similar numbered points in the profile $E^1\ D^1$, thus obtaining the points of intersection D^v, 5^v, 6^v, 7^v and E^v. Trace a curved line through these points, which will give the the top view of ED. As the radial lines drawn through the points $5'$, $6'$ and $7'$ on the line ED of the smaller cone in elevation intersect the section lines $b e$, $f k$ and $l n$ respectively, the radial lines in plan drawn through the apex L^1 and points 5^v, 6^v, and 7^v must intersect similar section lines in plan $b' e'$, $f' k'$ and $l' n'$ respectively, as shown by points 5^x, 6^x and 7^x. The points a' and o' are obtained by dropping perpendiculars from the points a and o in elevation onto the line $E^1 F^1$ in plan. Through the points thus obtained, draw the curved line a', 5^x, 6^x, 7^x, o' which will represent the plan view of one-half of the intersection between the two cones, the other half being similar.

Now from the intersections 5^x, 6^x and 7^x on the section lines $b' e'$, $f'' k'$ and $l' n'$ respectively, erect perpendicular lines intersecting similar section lines in elevation $b e$, $f k$ and $l n$ as shown respectively by points 5°, 6° and 7°.

A curved line traced through a, 5°, 6°, 7° and o will represent the line of intersection between the two cones in elevation. At right angles to the axis of the smaller cone and from the intersections a, 5°, 6° and 7° draw lines intersecting the side of the cone $E o$ at D^A 5^A 6^A and 7^A. For the pattern of the smaller cone proceed as is shown in the following plate:

PLATE III.

On this plate the two patterns should be placed in the center of the sheet. Take the radius of LD in Plate II and with L in Fig. 1 of Plate III as center describe the arc DD. From L drop a vertical line as shown by L E^A. Upon the arc DD measuring from either side of the point E, lay off the stretchout of the semi-circle E, 7, 6, 5, D in Plate II as shown by similar letters and figures on DD in Fig. 1 Plate III. From the apex L and through these points draw radial lines as shown and intersect them by arcs whose radii are equal to L D^A, L 5^A, L 6^A, L 7^A and L E^A in Plate II, as shown by similar letters and figures in Plate III. Trace a line through points thus obtained, and D, E, D, D^A, E^A, D^A, D will be the pattern for the small cone. As the pattern for the larger cone is obtained in the usual manner, we will only show how to obtain the opening to be cut into one side of the larger

PLATE III

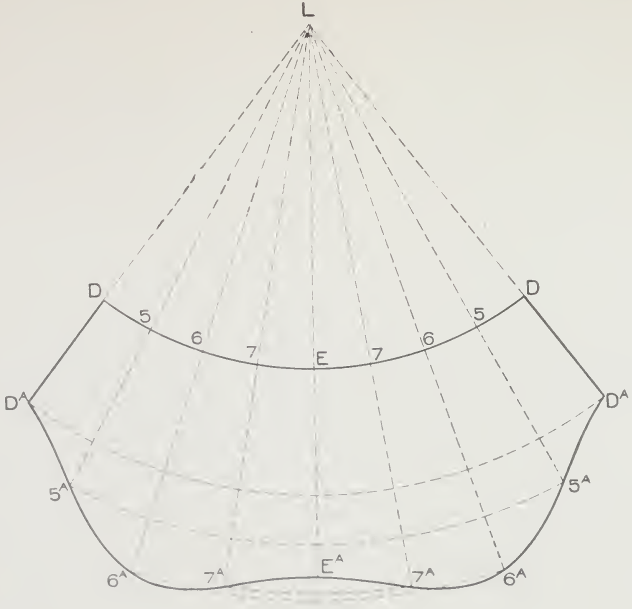


FIG. 1

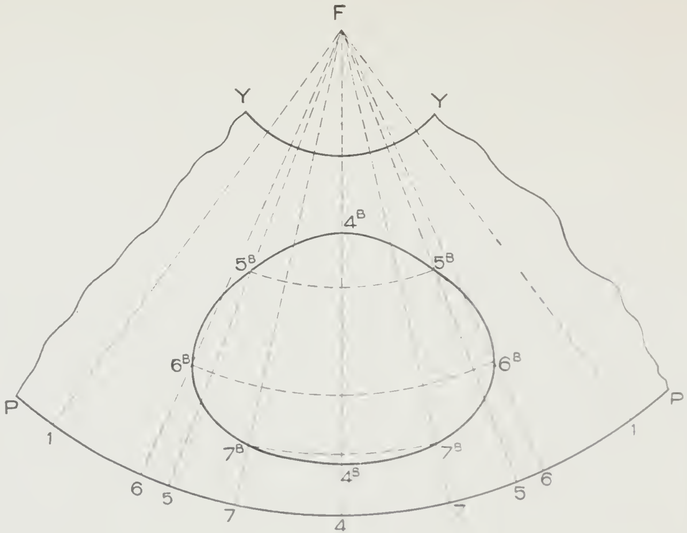
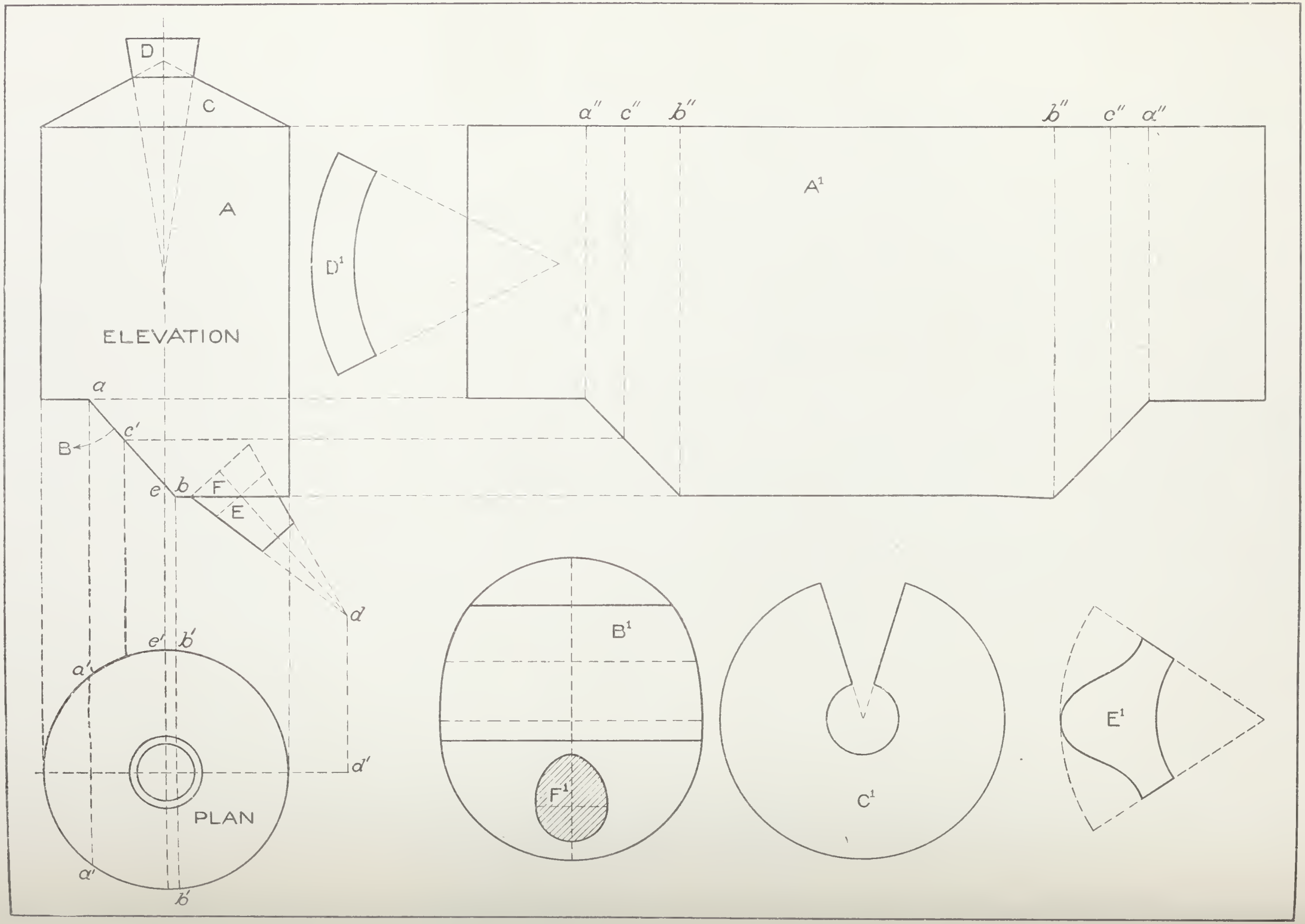


FIG. 2

PLATE IV



cone to admit the intersection of the smaller. We must now again refer to Plate II. From the intersections a, 5, 6, 7, and a in elevation draw lines at right angles to the line of the axis, intersecting the side of the cone at 4^B, 5^B, 6^B, 7^B and 4^B.

Also in addition to the spaces 1, 2, 3 and 4 in the plan view, it will be necessary to obtain the points of intersection on the base line in plan, where the radial lines would intersect drawn from the apex F^1 through the points of intersections between the two cones. This is accomplished by drawing lines from F^1 through 5^x, 6^x and 7^x until they intersect the base line in plan at 5, 6 and 7. All these points form the basis with which to develop the pattern shown in Fig. 2 of Plate III, in which draw the vertical line F 4, and with F as a center and radii equal to FY, and F P in Plate II draw the arcs YY and PP in Fig. 2 of Plate III as shown. Now starting from the point 4 on the arc PP on either side, lay off the stretchout of 1, 6, 5, 7 and 4 in plan in Plate II as shown by similar numbers in Fig. 2 of Plate III. From the points 6, 5, 7 and 4 on either side draw radial lines to the apex F, which will be used to obtain the pattern for the opening. Now with F as center and radii equal to F 4^B, F 5^B, F 6^B, F 7^B and F 4^B in Plate II, describe arcs intersecting radial lines having similar numbers in Fig. 2 of Plate III as shown by intersections having similar numbers. A line traced through these points will be the required opening to be cut out of the pattern of the larger cone, one-half of which is shown by drawing radial lines from the points 1 and 1 to the apex F.

PLATE IV.

In drawing this plate, the same size paper and border lines should be used as for the preceding plates. The subject for this plate is an oil tank resting on inclined wooden racks. The problem involves patterns in parallel and radial-line developments. The drawing can be made to any convenient scale until the problems are understood and should be proven by paste-board models. It should be drawn to a convenient scale, placing the pattern to fill the sheet and make a neat appearance. The section, stretchout lines, construction lines, and developments should be numbered or lettered, so as to prove the thorough understanding of the problem, and then sent to the School for correction. The various parts in the elevation and patterns have similar letters. A represents the tank body, the pattern being shown by A^1. B shows the bottom, the pattern being shown by B^1. The cone top C and inlet D are shown developed by C^1 and D^1 respectively, while the outlet E and opening F in elevation are shown developed by E^1 and F^1 in the bottom B^1.

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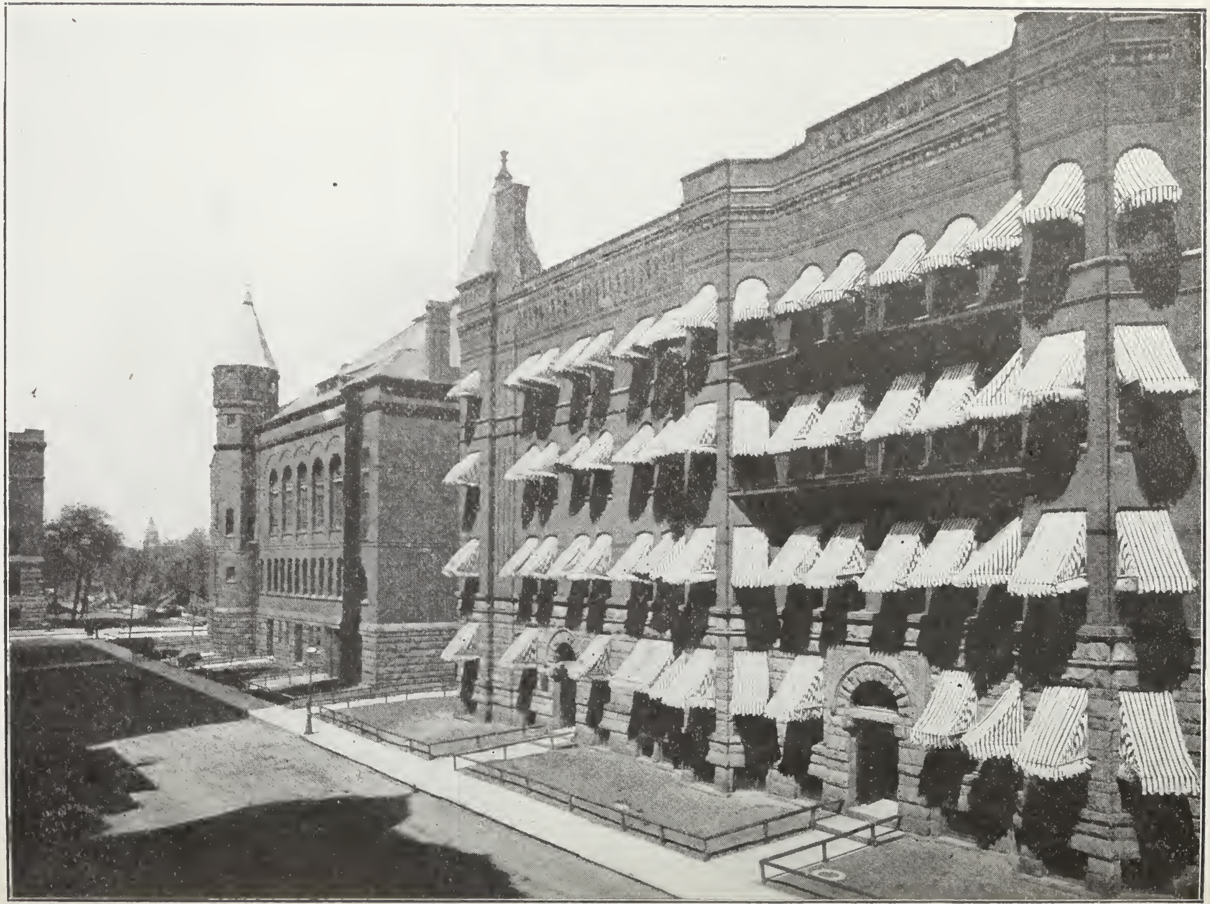
Machinery Hall, Armour Institute of Technology

THE FOLLOWING PAGES ARE TAKEN FROM THE BULLETIN OF THE AMERICAN SCHOOL OF CORRESPONDENCE AT ARMOUR INSTITUTE OF TECHNOLOGY, CHICAGO.

OTHER COURSES OFFERED ARE: ARCHITECTURE; HEATING, VENTILATING AND PLUMBING; CIVIL, ELECTRICAL, STATIONARY, LOCOMOTIVE, AND MARINE ENGINEERING; ELECTRIC WIRING; REFRIGERATION; TELEPHONY; TELEGRAPHY, TEXTILES, INCLUDING KNITTING; THE MAUFACTURE OF COTTON AND WOOLEN CLOTH, TEXTILE CHEMISTRY, DYEING, FINISHING, AND DESIGN; ALSO COLLEGE PREPARATORY, FITTING STUDENTS FOR ENTRANCE TO ENGINEERING COLLEGES.

THE COLLEGE PREPARATORY COURSE PRACTICALLY COVERS THE WORK OF THE SCIENTIFIC ACADEMY OF ARMOUR INSTITUTE OF TECHNOLOGY, AND IS ACCEPTED AS FULFILLING THE REQUIREMENTS FOR ENTRANCE TO THE COLLEGE OF ENGINEERING OF THAT INSTITUTION.

THE BULLETIN OF THE SCHOOL, GIVING COMPLETE SYNOPSIS OF THE ABOVE COURSES, MAY BE HAD ON REQUEST.



OFFICES, AMERICAN SCHOOL OF CORRESPONDENCE.

DEPARTMENT OF
MECHANICAL ENGINEERING

Partial list of the well-known authorities and educators who personally have prepared the instruction papers for correspondence study.

CHARLES L. GRIFFIN, S. B.,
Semet-Solvay Company.
EDWARD R. MARKHAM,
Rindge Manual Training School.
WALTER B. SNOW, S. B.,
B. F. Sturtevant Company.
WILLIAM T. McCLEMENT, A. M.,
Armour Institute of Technology.
JAMES RITCHEY,
Armour Institute of Technology.
WILLIAM NEUBECKER,
New York Trade School.

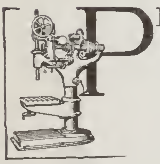
FREDERICK W. TURNER,
Mechanic Arts High School, Boston.
WALTER S. LELAND, S. B.,
Mass. Institute of Technology.
ARTHUR L. RICE, M. M. E.,
Editor, "The Engineer."
ERVIN KENISON, S. B.,
Mass. Institute of Technology.
RALPH H. SWEETSER, S. B.,
The Algoma Steel Co., Ltd.

COURSES

MECHANICAL ENGINEERING
MECHANICAL-ELECTRICAL ENGINEERING
SHEET METAL PATTERN DRAFTING
SHOP PRACTICE

BOILER MAKERS' COURSE
TOOL MAKERS' COURSE
SHEET METAL WORK
METAL ROOFING
CORNICE WORK
TINSMITHING

MECHANICAL ENGINEERING



PROBABLY no other branch of human knowledge has made such rapid progress during the last century as has Mechanical Engineering. Evidences of this are found in the improvement in the iron and steel industries, in agricultural implements, in textile machinery and in the creation of hundreds of new industries. The field of Mechanical Engineering is so broad that no one can expect to become expert in its whole extent. It includes steam engineering, the practical management of electrical machinery and the mechanic arts—pattern making, forging, foundry and machine shop work, tool making and sheet-metal work. This necessitates a division of the instruction; in this department special attention is given to the mechanic arts, leaving steam engineering to another department.

The general preliminary work—mathematics, the principles of machines, elementary chemistry and metallurgy—are the same in this course as in other branches of engineering. In addition to instruction in shop practice, the details of boilers and engines are discussed so that the mechanic working on steam machinery may know the relation of the parts to each other and to the whole machine.

In the Sheet-Metal Work, methods and principles are taught in a very thorough manner so that the student may take up new problems with little difficulty. As the laying out of patterns requires a working knowledge of intersections and developments, this portion of the course follows the study of projections in Mechanical Drawing.

MECHANICAL ENGINEERING COURSE

Prepared for Machinists, Toolmakers, Patternmakers, Moulders, Blacksmiths, Foundrymen, Builders of Boilers, Engines and General Machinery, Designers, Draftsmen, Engineers, Contractors, Superintendents and Foremen of Shops and Foundries.

INSTRUCTION PAPERS IN THE COURSE

Arithmetic Part I.
Arithmetic Part II.
Arithmetic Part III.
Elementary Algebra and Mensuration.
Mechanics Part I.
Mechanics Part II.
Mechanism.
Heat.
Elements of Chemistry.
Metallurgy.
Construction of Boilers.
Types of Boilers.
Boiler Accessories.
Pattern Making Part I.
Pattern Making Part II.
Foundry Work.
Forging.
Machine Design Part I.
Machine Design Part II.
Steam Pumps.
Steam Engine Part I.
Indicators.
*Gas Engines.
Valve Gears.
Steam Engine Part II.
Machine Shop Work Part I.
Machine Shop Work Part II.
Machine Shop Work Part III.
Tool Making Part I.
Tool Making Part II.
Tool Making Part III.
*Algebra Part I.
*Algebra Part II.
*Geometry.
*Trigonometry and Logarithms.
Mechanical Drawing Part I.
Mechanical Drawing Part II.
Mechanical Drawing Part III.
Mechanical Drawing Part IV.
Mechanical Drawing Part V.
Mechanical Drawing Part VI.
*Advanced Machine Design Part I.
*Advanced Machine Design Part II.
*Sheet Metal Work Part I.
*Sheet Metal Work Part II.

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*Optional.

SYNOPSIS OF COURSE

ARITHMETIC

FUNDAMENTAL PROCESSES: Units; Numbers: Concrete and Abstract, Like and Unlike. Notation; Numeration; Value; Arabic and Roman Numerals; Addition; Subtraction; Multiplication; Division; Factoring; Cancellation.

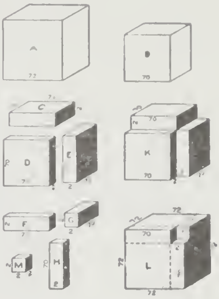
FRACTIONS: Definitions; Proper and Improper Fractions; Reduction of Fractions; Least Common Multiple; Least Common Denominator; Addition, Subtraction; Multiplication and Division of Fractions.

DECIMALS: Definitions; System of Notation; Addition, Subtraction, Multiplication and Division of Decimals; Symbols of Aggregation.

PERCENTAGE: Definitions; Decimals; Fractions and Percentage. Percentage; Rate; Base. Gain or Loss; Rules.

DENOMINATE NUMBERS: Reduction of Denominate Numbers; Linear Measure; Square Measure; Cubic Measure; Measures of Weight: Avoirdupois; Troy; Apothecaries'; Surveyors' Measure; Liquid Measure; Dry Measure; Measures of Money; Measures of Time. Operations with Denominate Numbers.

INVOLUTION AND EVOLUTION: Exponents; Powers; Radical Sign; Square Root; Cube Root. Roots of Fractions; Roots other than Square and Cube.



RATIO AND PROPORTION: Antecedent; Consequent; Extremes and Means; Rules for Solving Examples in Simple Proportion.

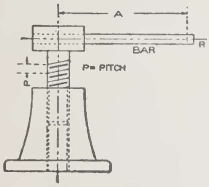
ELEMENTARY ALGEBRA AND MENSURATION

ALGEBRA: Use of Letters; Addition; Subtraction; Multiplication; Division. Cancellation; Simple Equations; Transposition; Substituting in Equations and Formulas; Finding Values of Unknown Quantities.

MENSURATION: Definitions: Lines; Angles; Polygons; Circles; Sectors; Segments; Measurement of Angles, Triangles, Rectangles, Trapezoids, Hexagons, Circles. Volumes and Surfaces of Prisms, Cylinders, Pyramids, Cones, Frustums of Cones and Pyramids. Sphere. Practical Problem: Measurement of Steam Space in a Horizontal Multitubular Boiler.

MECHANICS PART I

PROPERTIES OF MATTER: Atoms and Molecules; Solids; Liquids; Gases; Extensibility; Impenetrability; Indestructibility; Inertia; Divisibility; Porosity; Hardness; Tenacity; Brittleness; Malleability; Ductility.



MOTION, VELOCITY AND FORCE: Momentum; Newton's Laws; Parallelogram of Forces; Force Diagrams. Center of Gravity; Falling Bodies; Projectiles; The Pendulum; Kinetic and Potential Energy; Centrifugal Force.

PRINCIPLES OF MACHINES: Levers; Pulleys; Inclined Planes; Wedges; Screws; Laws of Friction; Coefficients of Friction. Tooth Gears: Spur; Worm; Bevel; Helical; Belt, Wire and Rope Gearing. Velocity Ratio; Horse Power Transmitted; etc.

MECHANICS PART II

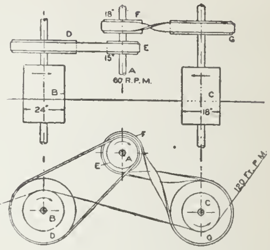
STRENGTH OF MATERIALS: Cohesion; Adhesion; Capillarity; Stress; Deformation; Elastic Limit; Breaking Strength; Coefficient of Elasticity; Tension; Compression; Shear; Torsion; Factor of Safety; Working Stress. Strength of Pipes and Cylinders; Strength of Beams and Columns; Moment of Inertia; Diagrams and Formulas.

HYDROSTATICS: Density; Specific Gravity; Hydrometers; Pressure of Fluids; Hydraulic Press; Head; Pressure; Dams and Gates; Coefficients of Flow; Water Motors; Turbines.

PNEUMATICS: Barometers; Mariotte's Law; Suction Pumps.

MECHANISM

REVOLVING BODIES: Mechanism; Motion; Velocity; Surface Speed; Calculation for Diameter and Number of Revolutions; Cylinder and Cones in Direct Contact. Cylinders and Cones Connected by Belts; Stepped and Tapered Cones. Disc and Roller. Tight and Loose Pulleys; Clutches; Other Mechanisms.

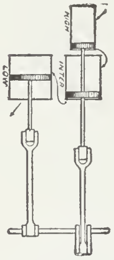


SIMPLE MACHINE PARTS: Screws; Levers; Cams; Linkwork; Motion and Power; Applications for Machine Shop and Textile Work. Quick Return Motions: Whitworth; Swinging Block.

GEARS: Spur; Annular; Bevel; Worm and Wheel; Velocity Ratios; Trains of Gears.

HEAT

DEFINITION AND NOTATION: Definition of Heat; Amount of Heat; Degree of Heat; Thermometer: Temperature; Fahrenheit; Centigrade; Reaumur. Freezing and Boiling Points. Notation: Absolute Temperature; Changing from one Scale to another.



HEAT EFFECTS: Expansion: Cubical; Linear; Coefficients of Expansion; Expansion of Solids, Liquids and Gases. Liquefaction: Laws of Fusion; Table of Melting Points; Vaporization; Evaporation; Boiling; Table of Boiling Points; Boiling under Pressure and in Vacuum. Distillation; Conduction; Convection; Radiation.

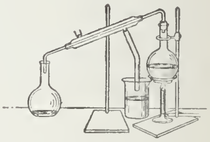
THERMODYNAMICS: Mechanical Equivalent of Heat; First Law; Adiabatic and Isothermal Expansion; Second Law; The Heat Engine.

PRACTICAL APPLICATIONS: The Steam Engine and Hot-Air Engine; Manufacture of Ice; Production of Liquid Air.

ELEMENTS OF CHEMISTRY

FUNDAMENTAL PRINCIPLES: Physical and Chemical Changes; Mechanical Mixture; Molecules and Atoms.

ELEMENTS: Table of Elements with Symbols, etc. Atomic Weight; Vapor Density; Law of Definite Weight; Valence; Combining Value. Equations, Factors and Products. Saturated Solution.

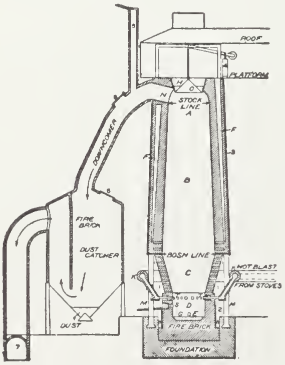


PROPERTIES: Physical and Chemical Properties of Oxygen; Hydrogen; Nitrogen. Chemistry of Air and Water; Carbon; Forms and Compounds. Illuminating Gas; Water Gas; Coal Gas; Composition. Combustion; Carbon Monoxide; Carbon Dioxide.

METALS: Sodium: Properties; Compounds. Calcium: Properties; Compounds.

ACIDS: Manufacture of Sulphuric, Nitric and Hydrochloric Acids.

METALLURGY



IRON AND STEEL: Ores: Magnetite; Hematite; Limonite; Siderite. Minerals: Iron Pyrites; Franklinite. Preparation of Ores.

BLAST FURNACE: Ores; Flux; Fuel; Slag. Operation and Chemistry of Furnace. Cast Iron: Grades and Impurities. Malleable Cast Iron.

WROUGHT IRON: Direct Process; Puddling Process; Operation of Furnace. Welding.

STEEL: Blister Steel; Crucible Steel; Bessemer Process: Converter; Acid and Basic Bessemer; Cupola Process; Molds and Molding. Rails. Open Hearth Process: Martin Process; Siemens Process; Combination; Gas Producers. Special Steels; Casehardening; Effects of Elements in Steel; Tempering: Colors and Temperature.

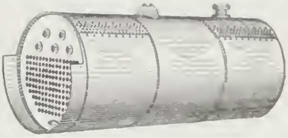
OTHER METALS: Copper: Ores; Extraction; Smelting; Refining. Lead: Smelting; Softening and Desilverizing; Refining. Tin. Zinc. Aluminum. Tables of Properties of Metals and Alloys.

STEAM BOILERS

CONSTRUCTION OF BOILERS: Materials; Testing; Punching; Drilling; Riveting; Joints; Arrangement of Plates; Staying; Expanding Tubes; Furnace Flues; Calking. Horse Power; General Requirements; Grate Area; Draft; Tubes; Steam and Water Spaces; Heating Surface; Water Level; Strength of Boiler Shell; Formulas for Thickness and Allowable Pressure; Equations and Formulas for Riveted Joints. Flues, Tubes and Stays; Manholes; Brackets; Uptake; Chimney.

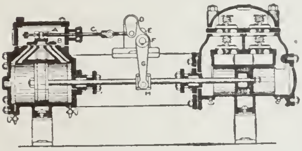
TYPES OF BOILERS: Cornish; Lancashire; Galloway; Multitubular; Vertical Boilers; Manning; Cochrane; Fire Engine; etc. Water-Tube Boilers: Babcock & Wilcox; Root; Heine; Almy; Cahill; Belleville; Yarrow; Thornycroft; Field. Return and Through Tube Marine Boilers; Locomotive Boilers.

BOILER ACCESSORIES: Furnaces; Grate; Smoke Prevention; Mechanical Stokers; Fuel Economizers; Natural and Forced Draft. Valves: Globe; Gate; Check; Safety. Blow-Out Apparatus; Gage Cocks; Gage Glasses; Pressure Gages; Fusible Plugs; Separators; Traps; Supports; Manholes; Pipes; Pumps; Heaters; Injectors; Horse Power; Calorimeters; Factors of Evaporation; Corrosion and Incrustation; Explosions. Fuels: Coal; Coke; Wood; Petroleum; Composition and Heat Value of Fuels. Care of Boilers: Fittings; Feed Water; Firing; Cleaning; Inspection; Testing.



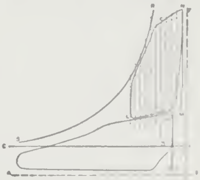
STEAM ENGINES

PUMPS: Principles of Action; Suction; Forcing; Jet; Rotary; Double Acting; Lift; Plunger; Geared; Duplex; Compound; Centrifugal. Valves: Arrangement of Parts; Capacity; Proportions; Sizes for Boiler Feed; Piping; Foundation; Packing; Lubrication; Speed; Running; Principal Types.



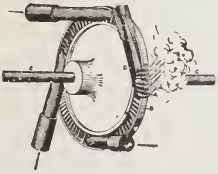
PARTS OF STEAM ENGINE: Cylinder; Piston; Crank; Crosshead; Valve; Eccentric. Types: Simple; Compound; Triple Expansion; Quadruple Expansion; High Speed; Pumping Accessories: Condensers; Cooling Towers; Fly Wheel; Governor; Lubrication; Steam Turbine; De Laval; Parsons; Advantages over Reciprocating Type.

INDICATORS: Diagrams; Mean Effective Pressure. Watts Indicator; Crosby, Thompson and Tabor Indicators; Reducing Motions; Taking Diagrams; Calculation of Horse Power. Table of Engine Constants; Measuring Area of Card; Planimeter; Indicator Cards; Actual; Ideal; Cards for Compound and Triple Expansion Engines; Combined Cards. Faults: Causes and Remedies; Calculation of Steam Consumption from Indicator Card.



VALVE GEARS: Slide Valve; Details; Operation; Eccentric; Valve Diagrams; Zeuner Diagram; Design of Slide Valve; Size of Parts; Setting; Equal Lead; Equal Cut-Off. Reversing Gears; Stephenson; Gooch; Allen; Joy Valve Gears. Other Valves: Piston; Double-Ported; Trick; Corliss. Methods of Balancing Valves.

ACTION OF HEAT: Gases; Boyle's Law; Saturated and Superheated Vapor; Efficiency of Actual Engine; Losses; Initial Condensation. Devices for Economy: Lagging; Jacketing; Superheating; Condensing; Measurement of Vacuum; Compounding; Corliss Valves; Separators. Piston Speed; Ratio of Expansion.



WORK DONE IN CYLINDER: Indicator; Formulas; Crank Action; Turning Moment. Testing: Indicators; Scales; Meters; Gages; Calorimeter Brake; Log of Test.

STEAM TURBINE: Steam Consumption, etc.

MACHINE DESIGN

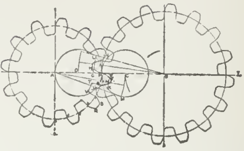
DEFINITION AND RELATION: Theoretical and Commercial Considerations; Invention; Original Design; Handbooks; General Reference Books; Empirical Data; Calculations; Notes; Records.

CONSTRUCTIVE MECHANICS AND DESIGN: Forces; Moments; Beams. Tension; Compression; Torsion; Friction and Lubrication. Working Stresses; Strains. Analysis of Conditions and Forces; Theoretical Construction; Practical Modification; Delineation and Specification.

APPLICATION TO POWER TRANSMISSION: Speed Ratio; Power; Load; Efficiency; Preliminary Calculation; Layout; Design of Parts; Belts; Pulleys; Shafts; Gears. Couplings; Clutches; Brakes. Bearings; Brackets; Stands; Bolts; Nuts; Screws; Keys; Pins; Cotters.

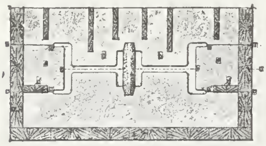
SIMPLE MACHINE ELEMENTS: Types; Materials; Shop Machinery; Processes; Construction; Cost; Competition; Machinery Market.

PLATES: Working Detail Drawings of Parts in a Practical Case of Power Transmission.

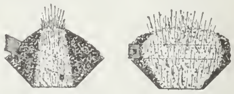


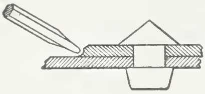
SHOP PRACTICE

PATTERN MAKING: Material; Kinds of Wood; Warping; Twisting. Tools; Saw; Plane; Chisel; Gouge; Square; Gauges; Compasses; Calipers. Machines; Trimmer; Grindstone; etc. Molding: Construction of Pattern; Working from Drawings; Shrinkage; Draft; Rappage; Simple Patterns. Bushing; Finishing Patterns; Shellac; Sandpaper; Gluing; Hand Screws; Pulley; Segments; Hand Wheel; Metal Patterns; Engine Crank; Disc Crank; Lathe Chuck; Large Cylinders; Engine Cylinders; Globe Valve; Gear Wheels; Templates; Patterns for Bevel Gears; Columns.



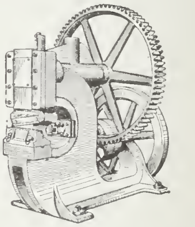
FOUNDRY WORK: Cupola: Charging; Blast; Flux; Preparation of Sand; Flasks; Loam; Binders. Tools: Shovel; Swab; Bellows; Rammer; Lifter; Smoother; etc. Molding: Molding Board; Drag; Pattern; Cope; Gating. Dry and Open Sand Molding; Loam Molding; Loose Pieces. Gates and Gating. Venting and Pouring: Vent Wires; Ladles; Daubing. Grades of Iron: Pig; Forge Irons. Castings: Cooling; Cleaning; Effect of Shape. Gaggers; Chaplets; Molding Columns; Gears; Pulleys; Pipes; Cylinders; etc.

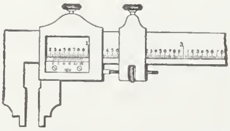




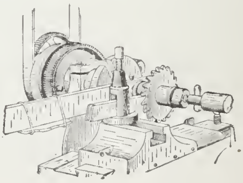
FORGING: Wrought Iron; Steel; The Forge: Tuyeres; Fuel; Blast. Tools: Hammer; Anvil; Tongs; Chisels; Flatter: Swages; Punches; etc. Forging and Finishing; Staples; Bolts; Hooks; Tongs; Wrenches; etc. Tool Dressing. Welding and Tempering: Temperature; Scale; Flux; Scarfing; Butt and Lap Welding; Welding Steel to Wrought Iron. Annealing; Case Hardening; Brazing. Power Tools: Hammers; Punches; Shears; Drop Forges; Bolt Headers; Presses. Forging Shafts, Connecting Rods; Cranks; etc. Shop Suggestions: Shrinkage; Expansion.

MACHINE SHOP WORK: Hand Tools; Hammer; Center Punch; Surface Gauge; Scales; Calipers; Micrometer; Vernier Micrometer; Gauges; Chisels; Files and Filing; Drills; Reamers; Taps and Dies. Lathes and Tools; Chucks; Dogs; Mandrels; Centering; Turning Tools; Turning; Tool Posts; Boring Tools; Cutting Speed; Grinding; Turning a Taper; Taper Attachment; Eccentric Turning. Boring. Boring Bars. Screw Cutting; Tools; Lead Screw; Gears; Compound Lathe; Chasing. Drilling in Lathe; Chucks. Drill Press; Drilling; Holding Work. Planer: Tools; Holding Work. Plate Planer. Shaper and Slotter. Milling Machine. Mills; Speed of Mills; Tables; Interchangeable Work. Grinding; Laying out Work; Shop Suggestions; Surface Plate; Drilling Hard Metals; Fitting Brasses; Fluting Rollers; Pickling; Lining up Shafting.





TOOL MAKING: Measuring Instruments; Annealing, Hardening, and Grinding Twist Drills. Reamers: Kinds; Cutting Edges; Straightening; Grinding; Adjustable Reamers; Reamer Holders. Expanding Mandrels; Eccentric Arbors; Milling Machine Arbors. Taps: Flutes; Hardening; Screw Die Hobs; Releasing Tap Holders. Screw Cutting Dies: Cutting Edges; Clearance; Spring Screw Threading Dies. Die Holders. Counterborers: Facing Tool with Inserted Cutter; Inserted Pilots; Combination Counterborers. Hollow Mills. Forming Tools; Holders. Milling Cutters: Teeth; Hardening and Grinding Hole to Size; Interlocking. Nicked and Inserted Teeth; End Mills; Spiral Mills; T-slot Cutters; Drill Jigs; Slab Jigs; Bushings; Legs; Leaf; Box Jigs. Punch and Die Work: Guide; Stripper; Gauge Pin; Shear. The Punch: Laying Out; Shear; Locating Pine; Bending Dies; Forming Dies. Gauges: Accuracy; Grinding; Lapping.



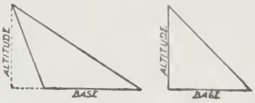
MATHEMATICS

ALGEBRA PART I

EXPRESSIONS: Symbols; Coefficients and Exponents; Symbols of Relation; Symbols of Abbreviation; Positive and Negative Terms; Monomial; Binomial; Trinomial; Polynomials; Similar Terms. Finding Numerical Value by Substitution. Finding Values of Unknown Quantities.

FUNDAMENTAL PROCESSES: Addition; Subtraction; Use of Parentheses; Multiplication; Division; Formulae; Factoring; Highest Common Factor; Lowest Common Multiple.

FRACTIONS: Fractions and Integers; Reduction of Fractions to Lowest Terms; Reduction of Fractions to Entire or Mixed Quantities; Reduction of Mixed Quantities to Fractions; Reduction of Fractions to Lowest Common Denominator; Addition and Subtraction of Fractions; Multiplication and Division of Fractions; Complex Fractions.



ALGEBRA PART II

SIMPLE EQUATIONS: Transposition; Solution of Simple Equations; Solution of Equations Containing Fractions; Literal Equations; Equations Involving Decimals; Equations Containing Two Unknown Quantities: Elimination by Addition, Subtraction, Substitution and Comparison.

INVOLUTION AND EVOLUTION: Monomials and Polynomials; Squares, Cubes and Higher Powers. The Radical Sign; Theory of Exponents; Radicals; Reduction of Radicals to Simplest Form; Addition, Subtraction, Multiplication and Division of Radicals. Involution and Evolution of Radicals. Irrational Denominators; Approximate Values.

IMAGINARY QUANTITIES: Multiplication and Division of Imaginary Quantities. Quadratic Surds.

EQUATIONS: Solution of Equations Containing Radicals. Pure and Affected Quadratic Equations; Simultaneous Equations Involving Quadratics.

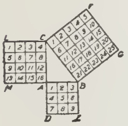
RATIO AND PROPORTION: Alternation; Inversion; Composition; Division.

PROGRESSION: Arithmetical and Geometrical.

BINOMIAL THEOREM: Formulae; Positive Integers; Finding Terms in an Expansion.

GEOMETRY

DEFINITIONS: Principles; Axioms; Abbreviations. Angles: Acute; Obtuse Complementary; Supplementary; etc. Parallel Lines; Axioms.



FUNDAMENTAL THEOREMS: Plane Figures; Polygons: Equilateral and Equiangular. Quadrilaterals; Circles; Measurements of Angles; Similar Figures; Trapezium; Trapezoid; Parallelogram; Rectangle; Square; Rhomboid; Rhombus. Ratio and Proportion. Terms; Alternation; Inversion; Composition and Division. The Circle: Theorems; Area; Circumference, etc.

SIMILAR POLYGONS: Definitions. Theorems. Areas of Miscellaneous Figures; Equivalent Polygons: Rectangles, Parallelograms.

PROBLEMS OF CONSTRUCTION: Twenty-nine Problems in Construction of Plane Figures.

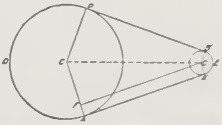
TRIGONOMETRY AND LOGARITHMS

TRIGONOMETRY: Definitions; Functions of Acute Angles; Measurement of Angles; Complementary Functions. Fundamental Theorems; Theorems Connecting the Different Functions of an Angle.

FUNCTIONS: From One Function of an Angle to Find the Other Functions. Functions of 45 degrees, 30 degrees and 60 degrees. Trigonometric Functions of Any Angle. Positive and Negative Angles; The Four Quadrants. Functions of 0 degrees, 90 degrees, 180 degrees and 270 degrees. Angles and Triangles.

LOGARITHMS: Nature and Use of Logarithms; Logarithms of a Product, a Fraction, a Power, a Root. Solutions of Arithmetical Problems by Logarithms.

TRIANGLES: Right Triangles: Solution by Natural Functions; Solutions by Logarithms; Areas. Oblique Triangles: Solution by Breaking up into Right Triangles; Areas.



EXERCISES: Length of Belt over two Pulleys; Stress in Rods forming an Acute Angle.

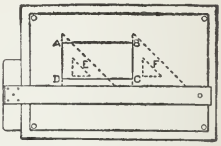
MECHANICAL DRAWING

PART I

INSTRUMENTS AND MATERIALS: Drawing Paper; Board; Pencils; T-Squares; Triangles; Compasses; Line Pens; Scales; Irregular Curves; Lettering Plates: Exercises in Line Work.

PART II

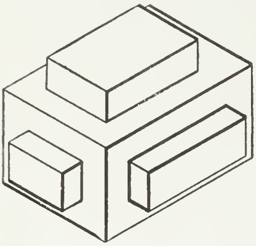
GEOMETRICAL DRAWING: Lines; Angles; Triangles; Quadrilaterals; Parallelograms; Rhombus; Pentagon; Hexagon; Circles; Measurement of Angles. Solids: Prisms; Pyramids; Cylinders; Cones; Frustrums; Spheres. Ellipse; Parabola; Hyperbola; Cycloid and Involute Curves. Plates: Twenty-Eight Problems in Geometrical Drawing.



PART III

PROJECTIONS: Orthographic: Plan and Elevation; Projection of Points, Lines, Surfaces and Solids. Third Plane of Projection; True Length; Shade Lines; Light and Dark Surfaces. Intersections and Developments: Intersections of Planes with Cones and Cylinders; Development of Prisms, Cylinders, Cones, etc. Development of Elbow. Isometric: Isometric Axes; Cube; Cylinder; Directions of Rays of Light. Plan and Elevation of Pentagonal Pyramid. Vertical and Horizontal Projections. Oblique Projections: Difference between Oblique Projection and Isometric. Shade Lines; Co-ordinates. Isometric of House, Box, etc. Oblique Projection of Crank Arm.

LINE SHADING AND LETTERING: Gradations of Light and Shade on Curved Surfaces; Shading Cylinders, Cones, Spheres, etc. Sizes and Spacing of Letters; Gothic and Roman Alphabets; Architectural Letters; Titles for Working Drawings. Plates.



PART IV

WORKING DRAWINGS: Lines: Full; Invisible; Shade; Center; Extension; Dimension. Location of Views; Cross Sections; Crosshatching; Dimensions; Finished Surfaces; Material; Conventional Representations of Screw Threads, Bolts and Nuts. Methods of Drawing Hexagonal and Square Nuts. Threads in Sectional Pieces; Broken Shafts, Columns, etc. Tables of Standard Screw Threads, Bolts and Nuts. Scale Drawing; Assembly Drawing; Blue Printing; Formulas for Blue-Print Paper. Plates. Plates.

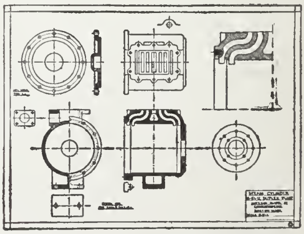
PART V

DETAILED DRAWINGS: The Helix; Pitch; Springs; Conventional Representations; V-Thread; Square Thread; Standard Threads. Cams: Kinds of Motion; Kinds of Cams; Designing.

GEARING: Belt: Parallel Shafts; Open and Crossed Belts; Quarter-Turn Belt; Reversible Quarter-Turn Belt with Two Guide Pulleys; With One Guide Pulley. Belts Connecting Non-Parallel Shafts whose Axes Intersect. Belt Holes. Tooth Gearing: Pitch Circles; Addendum; Back Lash; Diametrical Pitch; Cycloidal and Involute Gears; Epicycloidal Spur Gears. Annular Gears; Rack and Pinion; Involute Gears; Bevel Gears. Plates.

PART VI

WORKING SHOP DRAWINGS: Plan and Scope of Advanced Work; Essentials; Completeness of Drawings; Cost of Producing; Method of Procedure.



DUPLEX PUMP PLATES: Rating of Pump; Steam-End Layout; Molding and Machining of Steam End. Dimensions and Letters; System; Accuracy; Clearness; Completeness; Character; Inking and Tracing: Dimensions; Abbreviations; Piston Rod and Valve Stem; Molding and Machining; Steam Chest and Valve; Valve-Motion Layout and Details; Yoke; Stuffing Boxes; Brackets; Water-end Layout; Water Cylinder; Cap; Air Chamber; Plunger; Valve Details; Foundation; General Drawing.

ORDER SHEETS: List of Castings and Miscellaneous Parts: Bolts; Nuts; Pins; Special Fittings; Wrenches.

PLATES: Details; Pencil Drawings and Inked Plates; Assembly Drawing of Pump: Tracing.

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A complete practical course in Sheet Metal Work and Drafting prepared for all classes of Sheet-Metal Workers, Roofers, Contractors, Tinsmiths, Plumbers, etc.

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- Mechanics Part II.
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- Heat.
- Elements of Chemistry.
- Metallurgy.
- Mechanical Drawing Part I.
- Mechanical Drawing Part II.
- Mechanical Drawing Part III
- Tinsmithing.
- Sheet Metal Work Part I.
- Sheet Metal Work Part II.
- Metal Roofing.
- Cornice Work.
- *Algebra Part I.
- *Algebra Part II.
- *Geometry.
- *Trigonometry and Logarithms.
- *Mechanical Drawing Part IV.
- *Mechanical Drawing Part V.
- *Mechanical Drawing Part VI.

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A short, practical course for Sheet-Metal Workers, Tinsmiths, Plumbers, Carpenters, Contractors and others.

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- Arithmetic Part II.
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- *Metallurgy.
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- Mechanical Drawing Part II.
- Mechanical Drawing Part III.
- Sheet Metal Work Part I.
- Sheet Metal Work Part II.

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- *Heat.
- *Metallurgy.
- Mechanical Drawing Part I.
- Mechanical Drawing Part II.
- Mechanical Drawing Part III.
- Tinsmithing.

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*Optional.

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*Heat.
*Metallurgy.
Mechanical Drawing Part I.
Mechanical Drawing Part II.
Mechanical Drawing Part III.
Metal Roofing.

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*Optional.

CORNICE WORK

A short, practical course prepared for Sheet-Metal Workers, Tinsmiths, Roofers, Architects, Contractors and others.

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Arithmetic Part I.
Arithmetic Part II.
Arithmetic Part III.
Elementary Algebra and Mensuration.
*Heat.
*Metallurgy.
Mechanical Drawing Part I.
Mechanical Drawing Part II.
Mechanical Drawing Part III.
Cornice Work.

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A short course prepared for Boiler Makers, including instruction in Arithmetic, Mensuration and Mechanical Drawing.

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Mechanical Drawing Part I.
Mechanical Drawing Part II.
Mechanical Drawing Part III.
Mechanical Drawing Part IV.
Sheet Metal Work Part I.
Sheet Metal Work Part II.
Construction of Boilers.
Types of Boilers.

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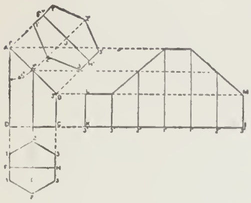
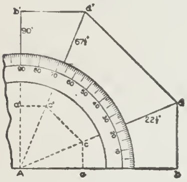
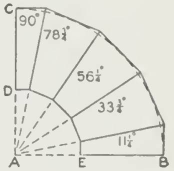
*Optional.

SYNOPSIS OF SHEET METAL COURSES

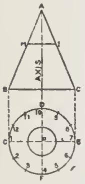


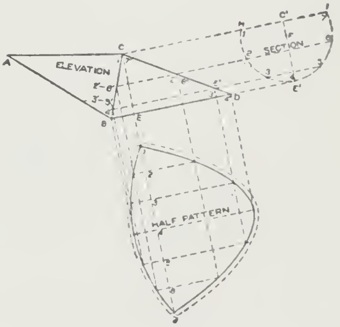
SHEET METAL WORK: Tables; Tools; Methods of Obtaining Patterns; Intersections and Developments. Approximate Developments. Circular Molds. Practical Shop Problems: Bathtub, Hip Bath, Funnel Strainer, Pail, Sink Drainer, Ventilator, Elbows in Round, Square and Oval Pipes, Tapering Elbows in Round and Square Pipes, Tapering Elbow Round to Oval, Rain-water Cut-off, Transition Piece in Rectangular Pipe, Curved Rectangular Chute, Hopper Register Box, Transition Piece from Round Base to Round Top placed Vertically. Offset Round to Oval. Threeway Branch Round to Round. Two-branch Fork Oval to Round. Tapering Flange Around Pipe Passing Through an Inclined Roof. Round Pipe Intersecting Conical Furnace Top; to One Side of Center of Top. Sphere, Ogee Basin, Brewer's Kettle, Moulded Cap on Stack, Elbows in Heavy Metal, Intersecting Pipes in Boiler Work, Gusset Sheet on Boiler, Scroll Signs in Heavy Metal.



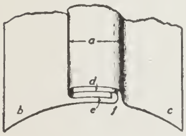


TINSMITHING: Principles and Methods of Construction; Tables; Capacity of Vessels; Shop Tools; Methods of Obtaining Patterns; Joints; Intersection and Developments: Prisms; Cylinders; Cones; Pyramids, Practical Tinware Problems: Short Rules; Allowance for Seaming and Wiring; Notching the Patterns; Transferring Patterns. Patterns for Pail, Funnel, Scoop, Pan, Coffee Pot, Foot Bath, Wash Boiler, Quart Measure, Scale Scoop, Dust Pan and Colander. Plates: Problems involving Intersections and Developments of Cones and Cylinders. Pattern for Oil Tank.





METAL ROOFING: Skylight Work: Tables; Tools; Patterns; Shapes of Bars and Curbs; Pitch; Hipped; Turrets; Conservatory; Photographic; Extension; Louvred. Ventilator; Curb; Hip Bar; Length of Bars. Coverings: Mensuration; Preparing Sheets; Laying; Base Flashings; Cap Flashings; Solder; Roof Flashing for Soil Pipes; Standing Seam Roofing: Locking Sheets; Preparing Roll; Bending; Cleats; Seaming; Finishing; Eaves and Ridge; Counter Flashing; Corrugated Roofing and Siding: Tables; Fastening Wood to Iron; Laying a Valley; Flashing Around Chimneys and Smokestack. End Wall Flashings. Ridge Rolls; Sidings and Awnings; Shapes of Flat Castings.



CORNICE WORK: Shop Tools; Molding; Miters; Molding on Plane and Curved Surfaces; Square Panel Miter; Triangular Panel Miter; Miters of Different Profiles; Molding Intersecting Pilaster. Gable Molding: Square Turret; Horizontal Returns at Top and Foot of Segmental Pediment. Gable Molding; Gutter Miters; Moldings with Different Projections; Star; Hip Ridge Miter; Eyebrow Dormer; Bay Window. Curved Moldings: Principles; Plane Flare; Cove Mold; Ogee Mold; Bead; Bottom of Circular Bay; Molding curved in Plan and in Elevation. Bull's-eye Window.

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Mechanical Drawing Part I.
Mechanical Drawing Part II.
Mechanical Drawing Part III.
Mechanical Drawing Part IV.
*Mechanical Drawing Part V.
*Mechanical Drawing Part VI.
Metallurgy.
Machine Shop Work Part I.
Machine Shop Work Part II.
Machine Shop Work Part III.
Toolmaking Part I.
Toolmaking Part II.
Toolmaking Part III.
Pattern Making Part I.
Pattern Making Part II.
Foundry Work.
Forging.

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*Optional.

Note.—Last five subjects sent in any order, as requested.

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*Elementary Algebra and Mensuration.
Metallurgy.
Forging.
Machine Shop Work Part I.
Machine Shop Work Part II.
Machine Shop Work Part III.
Toolmaking Part I.
Toolmaking Part II.
Toolmaking Part III.
*Mechanical Drawing Part I.
*Mechanical Drawing Part II.
*Mechanical Drawing Part III.
*Mechanical Drawing Part IV.
*Mechanical Drawing Part V.
*Mechanical Drawing Part VI.

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*Optional.

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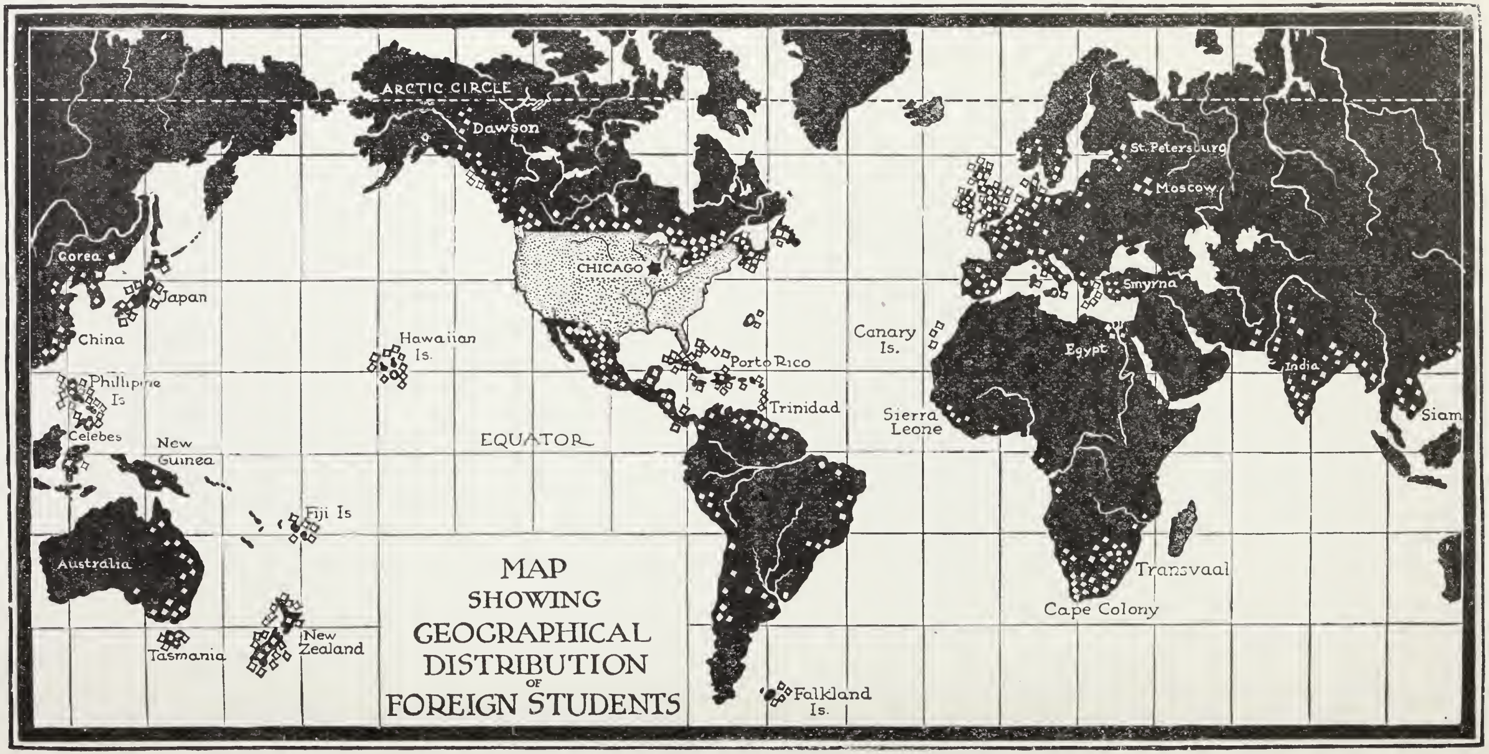

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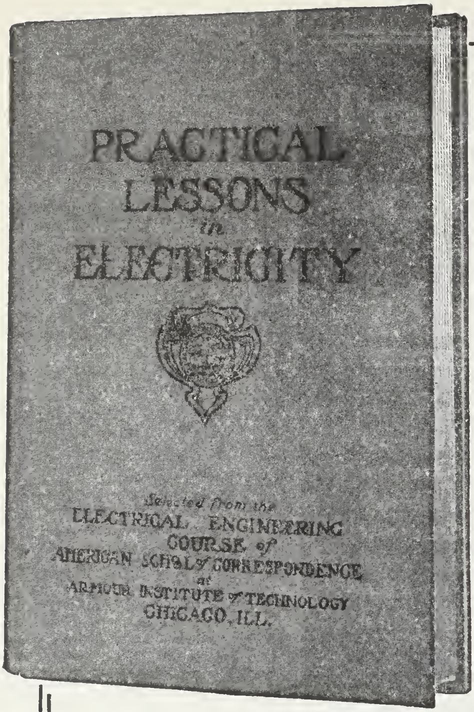

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